Comparative studies on engine behavior in a DI diesel engine fueled with UTO at different compression ratios

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THESIS SUBMITTED IN PARTIAL FULFILLMENT OF

THE REQUIREMENTS FOR THE DEGREE OF

Master of Technology

In

Mechanical Engineering

(Specialization: Thermal Engineering)

By

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CERTIFICATE

This is to certify that the thesis entitled, "**Comparative studies on engine behavior in a DI diesel engine fueled with UTO at different compression ratios**" submitted by **Mr. Sandip Belas Ekka** in partial fulfillment of the requirements for the award of Master of Technology in Mechanical Engineering with Thermal Engineering specialization during session 2012-2013 in the Department of Mechanical Engineering, National Institute of Technology, Rourkela.

It is an authentic work carried out by him under my supervision and guidance. To the best of my knowledge, the matter embodied in this thesis has not been submitted to any other University/Institute for the award of any Degree or Diploma.

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DATE: **SANDIP BELAS EKKA**

ABSTRACT

A preliminary investigation on the utilization of used transformer oil (UTO) as an alternative fuel for a single cylinder, four stroke diesel engine with optimum injection timing of 20° bTDC (before top dead centre) and injection pressure of 200 bar showed that the engine fueled with UTO gave a lower thermal efficiency and higher smoke emission than the diesel operation at full load. This is because of higher viscosity and poor volatility of UTO compared to diesel. The present investigation is aimed to examine the combustion, performance and emissions of a single cylinder, four stroke, direct injection diesel engine developing 4.4 kW at a rated speed of 1500 rpm, at optimum injection timing and optimum injection pressure of 230 bar, with lower compression ratios of 17:1 and 16:1, fueled with UTO by varying the clearance volume. At lower compression ratio, the engine exhibits a lower thermal efficiency and more smoke level. The engine behavior was also tested at higher compression ratio of 18.5:1. The effect of compression ratio on emission parameters of the engine fueled with UTO in comparison with diesel fuel operation is obtained. The optimum compression ratio was found to be 18.5:1. The results indicated that at higher compression ratio, there was an increase in brake thermal efficiency, NO emission and reduction in smoke. At this CR, the NO emission was found to be increased by about 2.8% and 32.1% respectively than that of the UTO and diesel. The smoke emission was found to decreased by about 3.6% and 10.4% respectively compared to that of UTO and diesel at full load. Further, the engine fueled with UTO was subjected to operate with optimum injection timing and optimum fuel nozzle opening pressure at different compression ratio (two lower and one higher CRs). The results were compared with diesel operation and presented in thesis. A comparison was made between the results obtained from these investigations (i.e. operating the engine fueled with UTO at different compression ratio only with optimum injection timing and, optimum injection timing and optimum nozzle opening pressure). With optimum injection timing and optimum nozzle opening pressure, there was 3.25% and 2.14% increase in brake thermal efficiency and NO emission and 6.52% reduction in smoke at higher CR. At CR 17:1 the results are closer to diesel operation with optimum injection timing and optimum nozzle opening pressure.

KEYWORDS- Combustion, Compression Ratio, Diesel Engine, Emission, Performance, UTO.

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NOMENCLATURE

Sr. No.	Short form	Full form
1	BSFC	Brake specific fuel consumption (kg/kW h)
2	BSEC	Brake specific energy consumption (kJ/kW h)
3	NO	Nitric oxide
4	CO	Carbon mono-oxide
5	EGT	Exhaust gas temperature
6	BP	Brake power (kW)
7	UTO	Used transformer oil
8	UBHC	Unburnt hydrocarbon
9	CO_2	Carbon dioxide
10	TFC	Total fuel Consumption
11	CA	Degrees crank angle
12	CR	Compression ratio
13	WPO	Waste plastic oil
14	HRR	Heat release rate
15	ROPR	Rate of pressure rise

CHAPTER 1

INTRODUCTION

1.1 General Introduction

Crude oil is the lifeblood of the modern world, which serves in all the sectors that includes transportation, agricultural, commercial and domestic and power generation. In the year 2009, the world consumed an estimated 84 to 85 million barrels of oil. There is a growing demand and cost of liquid fuel in every country. In the last six decades, India's energy consumption rate increases by 16 times because of fast rate of population growth [1]. At this rate, the fossil fuel will not be available for a long time as the gap between supply and demand increases large. With continuous use of petroleum products world is moving towards technological growth as well as environmental degradation.

In order to face the crisis of energy in the future and growing concern with the pollution, the substitutes of petroleum fuels is necessary. A large amount of crude oil is imported from the foreign countries; it is also a reason for the development of alternative fuel. The use of alternative fuels is the only possible solution, which can be obtained either from the renewable sources or non-conventional sources. As we know that, compression ignition (CI) engine is widely used in several applications; the search for alternative fuels for CI engines is very important [2]. Some of the fuel falls under this category are alcohol, vegetable oil, bio mass, bio gas, used oil etc. Few alternative fuels can be directly used without any modification in the fuel or engine, but some of them need little modification to obtain relevant properties like conventional fuel. An important equipment used for transmission and distribution of electrical energy is power or electrical transformer at different power stations and distribution stations. The extent to which the transformer oils are resilient to physico-chemical stresses induced by operating condition depends on the saturation or stability of the constituent substances. High stability leads to minimum sludge formation with time. Higher the compositional purity, higher the cooling function and the heat conduction capacity of the fuel [3]. Due to electric and cyclic thermal stresses transformer oil suffers from continuous deterioration and degradation because of loading and the climate conditions. This will affect the life and as well as performance of the electrical transformer. The expected service life of a transformer is about 40 years which strongly depends on the manufacturer, design, materials used, assembly quality, operating conditions and maintenance [4]. Day by day, more transformers are installed and the old transformer oils have to be scrapped, so it is difficult to estimate the annual disposed quantity of UTO with the statistics. To avoid the hazardousness and deterioration of the oil, it is essential to monitor the characteristics of transformer oil continuously. The preventive maintenance is done according to the utilization of the transformer oil. It is important that transformer oil is less biodegradable so there is some difficulty with disposal, it could be contaminate our soil and waterways. There are many transformers installed in high population areas and malls. By the effective utilization of UTO, the disposal problem in open land and the environmental problem can be solved. Recent literature reveals that in a CI engine, the UTO can be used as an alternative fuel after a proper treatment to obtain a suitable property [5].

1.2 Need of alternative fuels

1.2.1 Scarcity of crude oil

Transportation is the main reason that the any country currently depends heavily on crude oil. Most of this oil is imported from foreign countries and the major source of foreign crude oil is the Persian Gulf (this region provides one-fourth of the world's current consumption of oil and nearly two thirds of the world's oil reserves). Due to insufficient supply and distribution instability, reliance on this fossil fuel must end. Oil is a finite resource, which means that its supply is limited and cannot be reproduced. It took millions of years for these oil reserves to accumulate and they have been used in less than two hundred years. It is estimated that the current known reserves of oil on earth will only be able to supply total world demand for the next 40 years. When these reserves are completely exhausted we will have to use alternative fuel sources. In the short-term future, there are alternative reserves available. This oil shortage created havoc and illustrates the need to rely more on non-foreign sources of fuel. It was at that time the countries began research and development of alternative fuels.

1.2.2 Environmental damage

It is defined as the change or disturbance to the environment, which is not desirable. The emissions from vehicles damage the environment and contribute to air pollution. Several major environmental problems are caused by the use of fossil fuels.

1.2.3 Global warming

Global warming, also known as the "Greenhouse Effect", is caused by an accumulation of carbon dioxide (CO_2) emissions that do not leave the Earth's lower atmosphere. We are already at risk because the level of greenhouse gas already high. CO_2 is the gas responsible for keeping the earth's climate warm to a higher temperature by reducing outward radiations. However, an excess amount of CO_2 in the Earth's atmosphere is building up due to fossil fuel emissions that contain large quantities of CO_2.

- The natural process of heating and cooling of the earth's surface: the ultraviolet (UV) rays coming from the Sun hit the earth's surface and warm the earth. These UV rays, after hitting the Earth's surface, should then be bounced back up into the atmosphere.
- The greenhouse effect: The unnatural accumulation of CO_2 in the lower atmosphere is forming a thick blanket. The UV rays travelling from the Sun are able to penetrate the CO_2 blanket. The trapped UV rays are remaining near the earth's surface and this is reflected in the increased incidence of cancer among the world's population. The additional heat that is now being trapped near the earth's surface is also causing a general rise in global temperatures and a melting of million year old glaciers. As a result, weather patterns are being affected. A shifting of weather patterns will cause storms, heat waves and droughts that will lead to possible crop failures and famines. Tropical diseases will increase due to the increase in temperature. Rising ocean and lake levels will lead to coastal flooding.

1.2.4 Oil spill

An oil spill is a major environmental disaster and it occurs when an oil tanker transporting oil to shore from an ocean-based oil rig sinks and the oil cargo is released into the ocean. An oil spill can also occur when an oil rig, located in the ocean, accidentally leaks or explodes. It is also important to note that when an oil spill occurs on water it is extremely difficult to prevent the oil from spreading. It is also impossible to clean up an oil spill that occurs in water. Pollution to oceans is an extensive and countless so that fish, other valuable and beautiful forms of marine life, and coastal lines (where the oil eventually washes up) are disastrously affected. However, the oil that is released into the ocean from oil spill spreads very far and affects large areas of ocean life. Eventually, the oil reaches coastal land and destroys the animal and plant life that are part of the coastal ecosystem. Oil is a toxic and poisonous substance. It is not to be exposed directly to the environment because human, animal and plant life, plus the natural functioning of the atmosphere, are not adapted to deal with the chemical composition of fossil fuels.

1.2.5 Acid rain

This is an after-effect of the use of fossil fuels. When oil or coal is burnt through combustion devices, sulfur contained in the oil is converted into sulfur dioxide and emitted as a pollutant. Even though the sulfur content of fossil fuels is low, millions of tons of sulfur are released into the air each year due to the heavy use of fossil fuels. Acid rain occurs because, sulfur dioxide and nitrogen oxide are released through fossil fuel combustion. Due to complex chemical reactions that take place in the atmosphere, sulfur dioxide is transformed into sulfuric acid and nitrogen oxide is transformed into nitric acid. As well, fisheries and other animal life that create their habitat in or near water systems are affected because marine life cannot thrive in waters that are heavily acidic. This, in turn, affects other animals that rely on fish and water for sustenance.

1.2.6 Air pollution

Pollution can be defined as anything that causes a reduction in purity of air. Any substance in the air which is harmful for humans and the environment is known as an air pollutant. Most of the earth's air pollution is caused directly as a result of emissions from fossil fuels. The process of combustion (which occurs when a fossil fuel runs through an engine and is "burned" in order to create energy or heat) releases gases and minute particles into the air. When foreign particles are introduced into the air, it is obvious that the quality of air is affected.

1.2.7 Health threat of fossil fuel use

As known, fossil fuel provides a reliable energy for consumer there is a major risk associated with it. Carbon dioxide, produced from combustion of fossil fuels, is not a poisonous gas, but it is dangerous to the earth's natural climate system. Particulate matter is comprised of tiny particles that remain in the emissions of fossil fuels. Carbon dioxide (CO_2) and nitrogen oxide (NO_x) are poisonous gases that are dangerous to humans when inhaled. These gases are produced mainly from mobile source emissions (i.e. vehicle exhaust emissions). Children and the old people are the most susceptible to developing asthma and other respiratory illnesses as a result of exposure to fossil fuel emissions. PM is of extreme concern to human health.

The smoke, also called "soot", contains fine particles that are resulted from the chemical components as a result of combustion of fossil fuels particularly in diesel engines. The worst type of particulate matter comes from coal and from diesel fuel. Diesel fuel, the cheapest and crudest form of gasoline, is the most hazardous fuel because it emits 10 times more particulate matter per mile than conventional gasoline engines [6].

Particulate matter (PM) is essentially a mixture of solid particles and liquid droplets. They are dispersed through vehicle emissions and remain suspended at low levels, so that when we breathe, these particles enter our nose and mouth and become embedded in the deepest recesses of the lungs. Numerous scientific studies have proven that the particulate matter emitted from CI engine cause the following;

- Premature death
- Cancer
- Acute respiratory illnesses (asthma, chronic bronchitis, painful breathing)
- Shortness of breath
- Heart disease
- Lung damage

1.2.8 Solution to reduce emissions

Emissions from vehicle and industries cannot be fully stopped or avoided because; vehicle and industries are inevitable to civilization in the present day. But, the emissions can be controlled. One of the solution and to reduce the emissions is the use of alternative fuels. These alternative sources of fuel are currently being researched and developed.

1.3 Various alternative fuels

Various alternate desirable in the form of solid, liquid and gas are discussed in the subsequent subsections.

Table 1.1 Different types of alternative fuels

Type of sources	**Alternative fuels**		
	Solid	**Liquid**	**Gaseous**
Renewable	Wood Charcoal Municipal Waste	Ethanol Methanol Biodiesel	Hydrogen gas Biogas Producer gas
Non renewable			Natural gas CNG LPG

1.3.1 Solid alternative fuels

Solid fuel refers to various types of solid material that are used as fuels to produce energy and provide heating, usually released through combustion. Solid fuels are not much suitable for the internal combustion engine but many experimental engines have been built till now.

1.3.1.1 Wood

The first energy source discovered by the mankind is wood. Wood can be used in the form of cut logs, Wood chips and saw dust for domestic heating and to provide process heat in timber and furniture industries. It is bulky fuel, has low ash and sulphur content so it can be easily burnt and reduces the problem of acid rain. Wood chips can be used in all plant sizes. Chip quality generally depends on the type of wood used, the equipment used for making the chips, sorting techniques and moisture content. Dry chips can be stored, but moist chips start to compost or combust as it left too long period [3, 7-11].

1.3.1.2 Charcoal

Charcoal is one of the important solid alternative fuels and in some cases as an export product. It is produced through thermo-chemical transformation of biomass with the oxygen deficiency (pyrolysis). In the pyrolysis process, more than half of the energy in the wood is lost but charcoal produced from this process has an advantage for the user as more even and cleaner combustion than fuel wood. Charcoal has the potential to be used in medium speed engine. The fuel consumption is more with charcoal due to its lower heating value [12-14].

1.3.1.3 Municipal waste

Municipal refuse is messy to handle and has a low and variable energy content about one third of coal. It is dried, sorted and shredded and then it can be burnt to obtain heat and power. It is commonly known as trash or garbage a waste type consisting of everyday product used by the people. The technology associated municipal waste to energy is strongly depends upon the quality of waste to be treated and local conditions [15]

1.3.2 Liquid alternative fuels

1.3.2.1 Ethanol

Ethanol is an alcohol-based alternative fuel that is made by fermenting and distilling crops such as corn, barley or wheat. Ethanol can be blended with gasoline to increase octane levels and improve emissions quality. Ethanol can be made from ethylene or from fermentation of sugar and grains. The consumption of ethanol is approximately 51% higher than the diesel engine. With a higher compression ratio, an engine gives more power output and better fuel economy than a lower compression ratio engine. In general, engine fueled with ethanol gives marginally lower power output and torque output compared to diesel. It exhibits less toxicity, therefore fewer the standard pollutants. Due to manufacturing and processing requirement the cost of methanol is high [3, 16 and 17].

1.3.2.2 Methanol

Methanol, also known as wood alcohol, can be used as an alternative fuel in flexible fuel vehicles that are designed to run on M85, a blend of 85 percent methanol and 15 percent gasoline, but automakers are no longer manufacturing methanol-powered vehicles. Methanol and methanol blends have a higher octane rating than gasoline, which reduces the knocking in the engine and gives a higher engine efficiency. Methanol will become an important alternative fuel in the future, however, as a source of hydrogen needed to power fuel-cell vehicles. Attention should be made, because alcohol particularly methanol contain traces of water and corrosive organic impurities which can affect pipes, pumps and fuel tank of engine. Alcohol has a high octane rating, but lowers the calorific value than petrol. Methanol contains 25% less energy par gallon than ethanol and 50% less than petrol [3, 18 and 19].

1.3.2.3 Biodiesel

Biodiesel is an alternative fuel produced from vegetable oils and animal fats or algae by transesterification process. Vehicle engines can be converted to use either the blended form with petroleum diesel or even 100% in an unmodified engine. Biodiesel is proposed as a heating fuel in boilers. A byproduct of transesterification process is glycerol; 100 kg of glycerol is produced with production of 1 tonne of biodiesel. Biodiesel is safe, biodegradable, reduces air pollutants associated with vehicle emissions, such as particulate matter, carbon monoxide and hydrocarbons [20-23].

1.3.3 Gaseous alternative fuels

1.3.3.1 Natural gas

Natural gas is an alternative fuel that obtained from oil well naturally under considerable pressure and be used in production of natural gasoline. Natural gas main component are methane upto 65% and small amount of ethane and hydrocarbon. The composition of natural gas may vary with place and time. It burns clean and is widely available to people in many countries through utilities that provide natural gas to homes and businesses. The engine running with natural gas has a lean air fuel mixture, because of that it has many advantages such as higher efficiency, reduced tendency of knocking and less heat loss. When used in natural gas vehicles- cars and trucks with specially designed engines- natural gas produces far fewer harmful emissions than gasoline or diesel [3, 24].

1.3.3.2 CNG

The natural gas obtained from petroleum products can be compressed and then it is called compressed natural gas (CNG). A high pressure natural gas transfers from CNG fuel storage tank to the engine. A fuel management system (CNG conversion kits) is used to reduce the pressure of gas to the operating pressure of the engine. The CNG is injected into the engine in same manner as gasoline is injected into a gasoline-fueled engine. The engine functions the same way as a gasoline engine. These conversion kits contain auxiliary parts like the converter, mixer and other conversion parts required for conversion. When CNG is used in modified diesel engine, there is a marginal reduction in performance of the engine but improvements in engine emissions. The engines run on CNG have more engine life compared to diesel engine [3, 24 and 25].

1.3.3.3 LPG

In automobile engine two types of LPG are used one is propane and other one is butane. As an alternative fuel LPG has higher potential for IC engine. Propane is a byproduct of natural gas processing and crude oil refining. Already widely used as a fuel for cooking and heating, propane is also a popular alternative fuel for vehicles. Due to higher octane rating, the combustion of LPG is smoother hence improved antiknock characteristics as a result engine running is smooth. The liquid is converted into gas by the help of converter. Propane produces fewer emissions than gasoline, and there is also a highly developed infrastructure for propane transport, storage and distribution [3, 26-28].

1.3.3.4 Hydrogen

Hydrogen can be directly used in CI engine and can be mixed with gaseous fuel such as natural gas, CNG, LPG; biogas etc. in CI engines on dual fuel mode. When hydrogen is mixed with gasoline the combustion inside the cylinder takes place rapidly. Today, hydrogen is produced from fossil fuel mostly, but it can be produced from potentially available renewable resources like wind, solar and biomass. Therefore, hydrogen is not only clean; it is also a renewable energy source (no fear of its depletion). On the basis of lower heating value, 1 kg of hydrogen contains approximately same energy as 1 gallon of gasoline. The engines operated with hydrogen have less CO and HC emissions because of carbon present in the fuel. Most of the exhaust emission would be converted into water, N_2, and NO_x. Hydrogen is kept as a cryogenic liquid at very low temperature or as compressed gas in a pressure vessel. Hydrogen is very sensitive fuel thus handling with great care is required [3, 29-33].

1.3.3.5 Biogas

Biogas is produced by anaerobic digestion of an organic matter. Potential raw materials available on a large scale are; cow dung, municipal wastes, and plants specially grown for this purpose like water hyacinth, algae, and certain types of grasses. The main components of biogas are methane and carbon dioxide. It has a low calorific value and ignition quality but octane number is high. It can be produced in rural areas with readily available materials; this is the main advantage of biogas. Methane and carbon dioxide is the major component of biogas [3, 34 and 35].

1.3.3.6 Producer gas

The term "Producer Gas" refers to, gas produced in a gasifier to power cars with ordinary internal combustion engines. Producer gas has a low heating value because of air which contains high percentage of N_2. Producer gas can run an SI engine directly as a sole fuel, but it can be used in CI engine by lowering the compression ratio and installing the spark ignition system. Another possible way is duel fuel mode, where producer gas can be blend by 0-90% with gasoline. The engine efficiency operated with producer gas solely or duel fuel has a reduced power output. The heating value of producer gas and air mixture is about 2500 kJ/m^3 compared to gasoline it is very low. To increase the efficiency of the diesel engine fueled with producer gas by increasing heating value, amount of combustible mixture, compression ratio, ignition advances and engine speed (below 2500 rpm) [3, 36-38].

1.4 Waste to energy

Waste to energy is a potential method to produce useful energy by any of the methods such as direct combustion, pyrolysis, gasification, fermentation etc.

1.4.1 Plastic oil

Plastics are just long hydrocarbon chains. Energy from waste particle can be obtained by a catalyst or liquefying in the absence of oxygen by a tubular continuous reactor. The structure is reformed by rearranging the chains between carbon and hydrogen atoms with the help of catalysts to have a high fuel value. Plastic oil has a low heating value and sulphur content than that of diesel. The blend of plastic oil and diesel can be used directly without any modification in the diesel engine. It is reported that there is an increase in efficiency and higher emission with plastic oil and diesel blend [39-43].

1.4.2 Tyre pyrolysis oil

Pyrolysis of waste automobile tyres saves the environment and gives yields three valuable materials. Pyrolysis of 1 ton of waste tyres produces approximately 420 litres of pyrolysis oil, and also produces 150 kg of steel wire and 270 kg of carbon black. Pyrolysis gas can be condensed recovering pyrolysis oil. The recovered oil usually has specific gravity about 0.93, a sulphur content (1.1%) as well as the residual carbon content. This oil can be further filtered prior to be used mainly as heating oil. It is a type of light fuel oil or commonly named LFO. Its kinematic viscosity is 2.6 centi-Stokes (cSt) which makes it a non-viscous liquid. The fire point and flash point of tyre pyrolysis oil is close to diesel fuel. When a diesel engine fueled with blend of tyre pyrolysis oil and diesel gives marginally higher efficiency with an increase in fuel consumption. As the percentage of tyre pyrolysis oil is increases in fuel, the fuel consumption will be more. Tyre pyrolysis oil has a high content of sulphur and carbon, and therefore an increase in engine exhaust will occur.

Another by-product related to the pyrolysis of waste automobile tyres is the carbon black. The calorific value of the tyre carbon black is 26 MJ per kg which makes it good as a solid fuel. It will require further processing from powder into briquette form to make it suitable for combustion. Heat required for pyrolysis of tyres for every 1 ton of tyres, the fuel needed to complete the conversion of the tyres into oil and carbon black, is about 52 litres of diesel or 2,100 MJ of heat [44-46].

1.4.3 Used oil

This oil is typically available in large quantities, making collection necessary only. Used oil mainly includes brake fluids, transmission fluids, other hydraulic fluids, electrical insulating oils, motor oils, greases, emulsions, machine shop coolants, heating media, metalworking fluids, and refrigeration oils. The used oil is heated just below the smoking point. The strain in the heated oils is removed by filtering. The used oil can be used directly or its blends with diesel according to suitable conditions [47, 48].

CHAPTER 2

LITERATURE SURVEY

2.1 General

Several researchers have conducted experiments to study the combustion, performance and emission characteristics of a diesel engine with alternative fuels. Research works on such studies show that different kinds of alternative fuels viz. methyl esters of rapeseed oil methyl ester, palm oil methyl ester, corn oil, olive kernel oil, deccan hemp oil, jojoba oil, paradise oil, eucalyptus oil, poon oil, pongamia pinnata methyl ester, coconut oil have been used as investigated alternative fuels for diesel engine [49].

Wallace, W. et al [50] described the effect of compression ratio on engine behavior, mechanical execution, and development of an automatic, hydraulically-actuated piston that provides a practical method of obtaining a variable compression ratio engine. When applied to compression-ignition engines, an increase in output of 50% (0.5 to 0.75 bhp/cu in.) has been achieved without a corresponding increase in maximum combustion pressure. By providing a high compression ratio for starting and light load conditions, the engine has demonstrated substantial improvements in cold starting ability as well as improved potential for multifuel operation.

Raheman H. et al. [51] investigated the performance of Ricardo E6 engine using biodiesel obtained from mahua oil (B100) and its blend with high speed diesel (HSD) at varying compression ratio (CR), injection timing (IT) and engine loading (L). The brake specific fuel consumption (BSFC) and exhaust gas temperature (EGT) increased, whereas brake thermal efficiency (BTE) decreased with increase in the proportion of biodiesel in the blends at all compression ratios (18:1–20:1) and injection timings (35- 45^0 before TDC) tested. However, a reverse trend for these parameters was observed with increase in the CR and advancement of injection timing. The BSFC of B100 and its blends with high speed diesel reduced, whereas brake thermal efficiency and exhaust gas temperature increased with the increase in load (L) for the range of compression ratio and injection timing tested. The differences of BTEs between HSD and B100 were also not statistically significant at engine settings of ‘CR20IT40’ and ‘CR20IT45’. Thus, even B100 could be used on the Ricardo engine at these settings without affecting the performance obtained using HSD.

Laguitton et al. [52] studied the effect of CR on the emissions of a diesel engine when CR is reduced from 18.4 to 16, in a single cylinder. This was achieved by modifying the piston bowl while maintaining the production engine squish clearance. Investigations on the effect of injection timing were performed at a number of the key operating points and the corresponding pressure-time curves analysed to help explain the measured results. It was reported that, although there was a small CO and HC penalty, lowering the compression ratio or retarding the injection timing results reduction in NO_x and soot emissions.

Cayin and Gumas [53] investigated the influence of the CR, injection timing and injection pressure on the performance and emission of a DI diesel engine using biodiesel blended-diesel fuel. The tests were carriedout at three CRs 17:1, 18:1, and 19:1. It was reported that the BTE increased with increase in the CR while BSFC and BSEC decreased. For the all fuels tested, there was an increase in the NO emission, while the CO, HC emissions and the smoke opacity decrease with increase in the CR.

Celik M.B. et al. [54] concluded that the methanol has a greater resistance to knock and it emits lower emissions than gasoline. As single cylinder small engines have low compression ratio (CR), and they run with marginally rich mixture, their power are low and emission values are high. The performance can be increased at high CR, if these engines are run with fuels which have high octane number. In this study, methanol was used at high CR to increase performance and decrease emissions of a single-cylinder engine. Initially, the engine whose CR of 6:1 was tested with gasoline and methanol at full load and various speeds. Then, the CR was raised from 6:1 to 8:1 and 10:1, gradually. The knock was not observed at the CRs of 8:1 and 10:1 when using methanol while the knock was observed at the CR of 8:1 when using gasoline. The knock was determined from the cylinder pressure–crank angle curves. The results showed that some reductions were obtained in the CO, CO_2 and NO_x emissions without a major power loss when using methanol at the CR of 6:1. By increasing the CR from 6:1 to 10:1 with methanol, the engine power and the brake thermal efficiency increased by about 14% and 36%, respectively. Moreover, CO, CO_2 and NO_x emissions were reduced by about 37%, 30% and 22%, respectively.

Deore, Eknath R. et al. [55] conducted experiments on 3.75 kW diesel engine AV1 single cylinder water cooled, Kirloskar make tested for blends of diesel with ethanol. Tests were conducted for three different compression ratios. Engine test setup was developed with moving cylinder head for variation of compression ratio to perform investigations using these blends. The engine performance studies were conducted with rope break dynamometer setup. Parameters like speed of engine, fuel consumption and torque were measured at different loads for pure diesel for blends of diesel with ethanol at different compression ratio. Break Power, BSFC, BTE and heat balance were evaluated. Results were recorded for 5% to 20% ethanol in the blend and three different compression ratios.

Muralidharan, K. et al. [56] carriedout investigations to evaluate the performance, emission and combustion characteristics of a single cylinder, four stroke, variable compression ratio multi fuel engine when fueled with waste cooking oil methyl ester and its 20%, 40%, 60% and 80% blends with diesel (on a volume basis) were investigated and compared with standard diesel. The suitability of waste cooking oil methyl ester as a biofuel has been established in this study. Bio diesel produced from waste sun flower oil by transesterification process has been used in this study. Experiment has been conducted at a fixed engine speed of 1500 rpm, 50% load and at compression ratios of 18:1, 19:1, 20:1, 21:1 and 22:1. The impact of compression ratio on fuel consumption, combustion pressures and exhaust gas emissions has been investigated and presented. The optimum compression ratio at which the engine gave the best performance has been identified. The results indicate that the engine exhibits longer ignition delay, maximum rate of pressure rise, lower heat release rate and higher mass fraction burnt at higher compression ratio for waste cooking oil methyl ester when compared to that of diesel. The brake thermal efficiency at 50% load for waste cooking oil methyl ester blends and diesel has been calculated and the blend B40 was found to give maximum thermal efficiency. The blends when used as fuels, results in reduction of carbon monoxide, hydrocarbon and increase in nitrogen oxides emissions.

Muralidharan K. et al. [57] carriedout experiments to estimate the performance, emission and combustion characteristics of a single cylinder; four stroke variable compression ratio multi fuel engine fueled with waste cooking oil methyl ester and its blends with standard diesel. Studies have been conducted using the fuel blends of 20%, 40%, 60% and 80% biodiesel with standard diesel, with an engine speed of 1500 rpm, fixed compression ratio 21 and at different loading conditions. The performance parameters evaluated includes the brake thermal efficiency, specific fuel consumption, brake power, indicated mean effective pressure, mechanical efficiency and exhaust gas temperature. The exhaust gas emission was found to contain carbon monoxide, hydrocarbon, nitrogen oxides and carbon dioxide. The blend when used as a fuel, results in the reduction of carbon monoxide (CO), hydrocarbon (HC), carbon dioxide (CO_2) at the expense of nitrogen oxides emissions. It was found that the combustion characteristics of waste cooking oil methyl ester and its diesel blends closely followed those of standard diesel.

Debnath, B. K. et al. [58] presented an experimental work is set to explore the effect of compression ratio (CR) and injection timing (IT) on energy and exergy potential of a palm oil methyl ester (POME) run diesel engine. Experiments were carried out in a single cylinder, direct injection, water cooled, variable compression ratio diesel engine at a constant speed of 1500. rpm under a full load of 4.24. bar brake mean effective pressure (BMEP). The study involves four different CRs of 16, 17, 17.5 and 18; and three different injection timing of 20°, 23° and 28°BTDC. Here, the CR of 17.5 and injection timing of 23°BTDC were the standard ones. The energy analysis performed for the experimental data includes shaft power, energy input through fuel, output by cooling water and exhaust, uncounted loss per unit time. Simultaneously the effects of varying CR and injection timing on peak pressure, peak heat release rate, brake thermal efficiency and exhaust gas temperature were also studied. The exergy analysis also showed that with the increase of CR, the injection retardation and advancement increased the shaft availability and exergy efficiency, while it reduced the exergy destruction.

Kale, B. N. et al. [59] described the use of cotton seed vegetable oil (biodiesel) as a fuel for variable compression ratio CI engines. The study was based on the reports of about 40 scientists including (some manufacturers and agencies) between 1996 and 2010. The tests were conducted using different types of raw and refined vegetable oils. It was found that the experiments with raw vegetable oil as fuels did not show the satisfactory results. A majority of scientists mixed the transesterified vegetable oil or biodiesel oil with diesel with different proportions. However, the refined, chemically processed (transesterified) vegetable oil mixed with diesel can be used to run compression ignition engine for longer duration. It was reported that there was a marginal decrease in brake power and a slight increase in fuel consumption. However, the lubricant properties of the vegetable oil are better than diesel, which can help to increase the engine life. Moreover, the vegetable oil fuel is environment friendly, produces much lesser NO_x and HC and absolutely no SO_x and no increase in CO_2 at global level.

Rao GVNSR et al. [60], In order to find out optimum compression ratio experiments were carried out on a single cylinder, four stroke, variable compression ratio diesel engine. Tests were carried out at compression ratios of 13.2, 13.9, 14.8, 15.7, 16.9, 18.1 and 20.2. Results showed a significant improved performance and emission characteristics at a compression ratio 14.8. The compression ratios (CR) less than 14.8 and greater than 14.8 showed a drop in break thermal efficiency, rise in fuel consumption along with increase in smoke densities.

2.2 Alternative fuels from waste substances

Literature review reveals that oils such as waste lubricating oil (WLO/ULO), waste plastic oil (WPO) and tire pyrolysis oil (TPO) obtained from waste automobile tires, and waste plastics oil derived from plastics by pyrolysis (WPOP) have already been investigated for their effective utilization as alternative fuels in CI engines.

Mani, M. et al. [41] analyzed and compared the properties of WPOP with the petroleum products and found that it has properties similar to that of diesel. After the analysis of oil, WPOP was used as an alternate fuel in a DI diesel engine without any engine modification. The performance, emission and combustion characteristics of a single cylinder, four-stroke, air cooled DI diesel engine run with waste plastic oil were obtained and compared with diesel operation. The experimental results have showed a stable operation and comparable brake thermal efficiency for WPOP with that of diesel. The unburnt hydrocarbon emission from WPOP fueled engine was found to be higher by about 15% compared to that of diesel operation. The CO emission for WPOP was higher by about 5% than that of diesel at full load. Smoke reduced by about 40% for WPOP compared to that of at all loads.

Mani, M. et al. [61] used WPO as an alternative fuel in a small powered diesel engine. The influence of injection timing on the performance, emission and combustion characteristics of a single cylinder, four stroke, direct injection diesel engine was studied using WPO as a fuel, at four different injection timings (23°, 20°, 17° and 14° bTDC). In comparison with standard injection timing of 23° bTDC the retarded injection timing of 14° bTDC, the WPO fueled engine gave a reduction in oxides of nitrogen, carbon monoxide and unburned hydrocarbon emission by about 4.4%, 25% and 30% respectively at full load. The brake thermal efficiency was found to be higher by about 4% for WPO compared to that of diesel operation. It was also noticed that smoke emission was found to be higher by 35% for WPO than that of diesel.

Murugan, S. et al. [44, 62] conducted tests to evaluate the performance and emission characteristics of a single cylinder, DI diesel engine fueled with 10, 30 and 50% blends of tire pyrolysis oil (TPO) with diesel. TPO was derived from waste automobile tires through vacuum pyrolysis in 1 kg batch pyrolysis unit. Results indicated that the brake thermal efficiency of the engine fueled by TPO-diesel blends increased with increase in blend concentration and higher than diesel at full load. The NO, HC, CO and smoke emissions were found to be higher at higher loads due to high aromatic content and longer ignition delay. The cylinder peak pressure increased from 71.4 bar to 73.8 bar. The ignition delays were longer than diesel. 80% and 90% of distilled tire pyrolysis oil (DTPO) blended with 20% and 10% diesel respectively were also used as fuels in a four stroke, single cylinder, air cooled diesel engine without any engine modification. The performance, emission and combustion characteristics of a single cylinder, four stroke, air cooled, DI diesel engine running with the DTPO -diesel blends at higher concentrations were studied. The brake thermal efficiency was found to be lower by about 3%. NO_x emissions were found to be lower by about 18% and smoke emissions were higher by about 38% compared to that of diesel at full load.

Murugan, S. et al. [45] also carried out an experimental investigation to utilize crude tire pyrolysis oil (TPO) as an alternative fuel in a diesel engine. TPO was desulphurised and then distilled through vacuum distillation. Also, two distilled tire pyrolysis oil (DTPO)-diesel fuel (DF) blends at lower (20%DTPO) and higher concentrations (90%DTPO) were used as fuels in a four stroke, single cylinder, air cooled, diesel engine without any engine modification. The results were compared with diesel fuel (DF) operation. Results indicated that the engine can run with 90% DTPO and 10% diesel fuel.

Arpa, O. et al. [63] examined WLO as an alternative fuel in a single cylinder, four-stroke, air cooled, naturally aspirated direct injection diesel engine developing a maximum power of 10 kW at 2000 rpm. Results of the investigation showed that there was a marginal increase in the brake thermal efficiency was obtained for WLO. The brake specific fuel consumption for WLO was also marginally lower compared to that of diesel fuel.

Tajima, H. et al. [64] carriedout experimental investigation to utilize the ULO as an alternative fuel in generator plants. The combustion characteristics of a diesel engine were determined by observing the burning flames in the engine, while on a test run. The results were compared with heavy fuel oil. The ULO showed better ignition quality and 64.71% lower smoke emission compared to heavy fuel oil operation. But, a thick deposit of combustion products was noticed in the combustion chamber after a short run. It was suggested that a process is required to remove the additives from ULO, before utilising it as a fuel in diesel engines.

The aim of this study is to develop a way to utilize UTO in diesel engine. It is an attempt has been made to assess the combustion, performance and emission characteristics of a CI engine fueled with UTO with different CRs. The experimental results are recorded, analyzed and compared with those of diesel operation and presented in this paper.

CHAPTER 3

MATERIALS AND METHODS

3.1 Nature of Transformer Oil

Inhibited oil is formulated with hydro treated naphthenic base oil and an oxidation inhibitor to control sludge and deposit formation. It provides an extended service life compared to non-inhibited transformer oils. It has an excellent low-temperature properties and is noncorrosive to copper and copper alloys. This oil does not contain any polychlorinated biphenyls. Mineral oil, synthetic esters and silicon oils are traditionally used as transformer oils. Mineral transformer oil is composed of hydrocarbons of paraffinic, aromatic or naphthenic structure that are obtained by fractional distillation of crude petroleum. Synthetic oils are produced by substituting a chlorine atom for a hydrogen atom in hydrocarbon molecules. Apart from these, transformer oil is also produced from various vegetable oils, such as coconut oil, sunflower oil, soybean oil and castor oil [65, 66].

3.2 Physical properties and chemical composition of UTO

A comparison of the chemical composition between of UTO and diesel is shown in Table 3.1. The chemical composition of the UTO indicates that the fuel has carbon close to that of diesel fuel. The hydrogen present in the UTO is 1.5 times lesser than that of diesel. It is evident from the table that the UTO has considerable oxygen present in it. Due to the oxidized nature, the fuel may be helpful in better combustion of the fuel air mixture. The kinematic viscosity of the UTO is approximately 6 folds higher than that of UTO. Table 3.2 gives the comparison of the physical properties between UTO and diesel.

Table 3.1 Chemical composition of UTO and diesel [5]

Description	Diesel	UTO
C (%)	86.5	89.95
H (%)	13.2	9.19
N (%)	0.18	0.03
S (%)	0.3	0.35
O by difference (%)	0	0.44
C/H ratio	5.437	19.302
Carbon residue (%)	0.02	0.02

Table 3.2 Properties of UTO and diesel

Property	Test method	Diesel	UTO
Kinematic viscosity (cSt@ 27°C)	D-0445	2.4	13
Flash point (°C)	D0093-02A	76	150
Fire point (°C)		56	172
Pour point (°C)	D0097-05A	-16	-16.7
Density (kg/m3)	D-1298	860	890
Lower calorific value (kJ/kg)	D-4809	44800	39270
Sulfur content (%)	D129-00R05	0.05	0.02
Cetane number		40-55	43.6*[67]
Carbon residue (%)	D2500-05	0.01	0.02
T10 (°C)		210	320
T50 (°C)		230	340
T90 (°C)		260	370
T100 (°C)		350	360

3.3 Method to change compression ratio

The compression ratio of the engine can be varied by five methods that are described below;

3.3.1 Cylinder swept volume

The swept volume of the cylinder indicates how much air the piston displaces as it moves from BDC to TDC. Increasing the cylinder volume without making any other changes will increase the compression ratio because it enlarges the cylinder volume without increasing the combustion chamber volume. In other words, the piston will have to suck more air into the same amount of space.

3.3.2 Clearance volume

Clearance volume is determined by the distance from the cylinder block deck to the top of the piston flat (not counting any dishes or domes) when the piston is at TDC. In engines, the pistons don't come all the way up to the height of the deck. They can be anywhere from 0.003 to 0.020 inch below it. This amount is known as the piston deck height, and it affects compression ratio because it affects the volume of air in the combustion area when the piston is at TDC. If the piston deck height is inreased, then clearance volume is increased and the compression ratio is reduced. If the piston is closer to the deck, then the clearance volume is reduced and compression ratio is increased.

3.3.3 Piston dome or dish

It is associated with the piston geometry. Instead of a flat top surface it has a dome which looks like a dome stadium. Clearance volume does not take into account any pop-up domes or sunken-in dishes on the head of the piston. These configurations also increase or decrease volume in the combustion chamber and affect the compression ratio. For the purpose of calculating compression, the dish is preferred as a positive quantity because the dish adds volume to the cylinder hence reduces the compression ratio. Dome is just reverse of the dish, it subtracts volume from the cylinder and increases the compression ratio.

3.3.4 Combustion chamber

The volume of the combustion chambers is the final factor in determining the compression ratio. The larger the chamber, the more volume is added to the cylinder and the lower the compression ratio; smaller chambers reduce the volume and increase the compression ratio.

3.3.5 Head-gasket volume

Head-gasket volume is determined by the compressed thickness of the gasket. A thicker gasket adds volume and reduces compression; a thinner gasket reduces volume and increases compression.

In this experiment, the change in compression ratio was done by change in head gasket volume. Here is how to calculate the volume of the head gasket;

$$\text{Head Gasket Volume} = 0.786 \times (\text{Cylinder Bore})^2 \times \text{Compressed Thickness}$$

Table 3.3 Clearance volume and gasket volume for different compression ratios

Compression ratio	Total clearance volume (cm^3)	Head gasket volume (cm^3)
16:1	44.12	9.67
17:1	41.32	6.92
17.5:1	40.01	5.6
18.5:1	38.21	4.21

3.4 Engine specification

The test engine used in this investigation was a Kirolaskar TAF-1 single cylinder, four-stroke, air cooled, constant speed, direct injection diesel engine. The specifications of the engine are given in Table 3.4.

Table 3.4 Specifications of the test engine

Parameter	Value/dimension
Speed (rpm)	1500
Bore (mm)	87.5
Stroke (mm)	110
Rated brake power (kW@1500 rpm)	4.4
Compression ratio	16:1, 17:1, 17.5:1, 18.5:1
Nozzle opening pressure (bar)	230
Injection timing (°CA bTDC)	20

3.5 Description of the test engine

Fig. 3.1 shows the schematic diagram of the experimental setup. A control panel (1) to provide electrical load to the engine (2) was fitted with the electrical resistance dynamometer (3) known as alternator. A fuel tank (4) was connected to the engine for continuous fuel supply. There was included a burette (5) and a fuel sensor (6) with the fuel circuit to measure the fuel consumption and give input to the computer (7) through data acquisition system (8). A pressure transducer (9) was mounted on the engine head to measure the cylinder pressure with loads. The model of the Kistler pressure transducer was 6613A, which has an advantage of a good frequency response and linear operating range. A continuous circulation of air was maintained for cooling the transducer, by using fins to maintain the required temperature. A crank position sensor (10) was connected to the output shaft to record the crank angle.

Combustion parameters such as the peak pressure, time of occurrence of peak pressure, heat release rate and ignition delay were obtained with the help of software provided by legion brothers. Atmospheric air enters the intake manifold of the engine through an air filter and an air box (11). An air flow sensor (12) fitted with the air box gave the input for the air consumption to the data acquisition system. All the inputs such as air and fuel consumption, engine brake power, cylinder pressure and crank angle were recorded by the data acquisition system, stored in the computer and displayed in the monitor. A thermocouple in conjunction with a temperature indicator (13) was connected at the exhaust pipe to measure the temperature of the exhaust gas. An AVL DiGas444 exhaust gas analyzer (14) was used to examine the engine exhaust components like CO_2, CO, O_2, HC, and NO in percent. The smoke density of the exhaust was measured by the help of an AVL437 (15) diesel smoke meter.

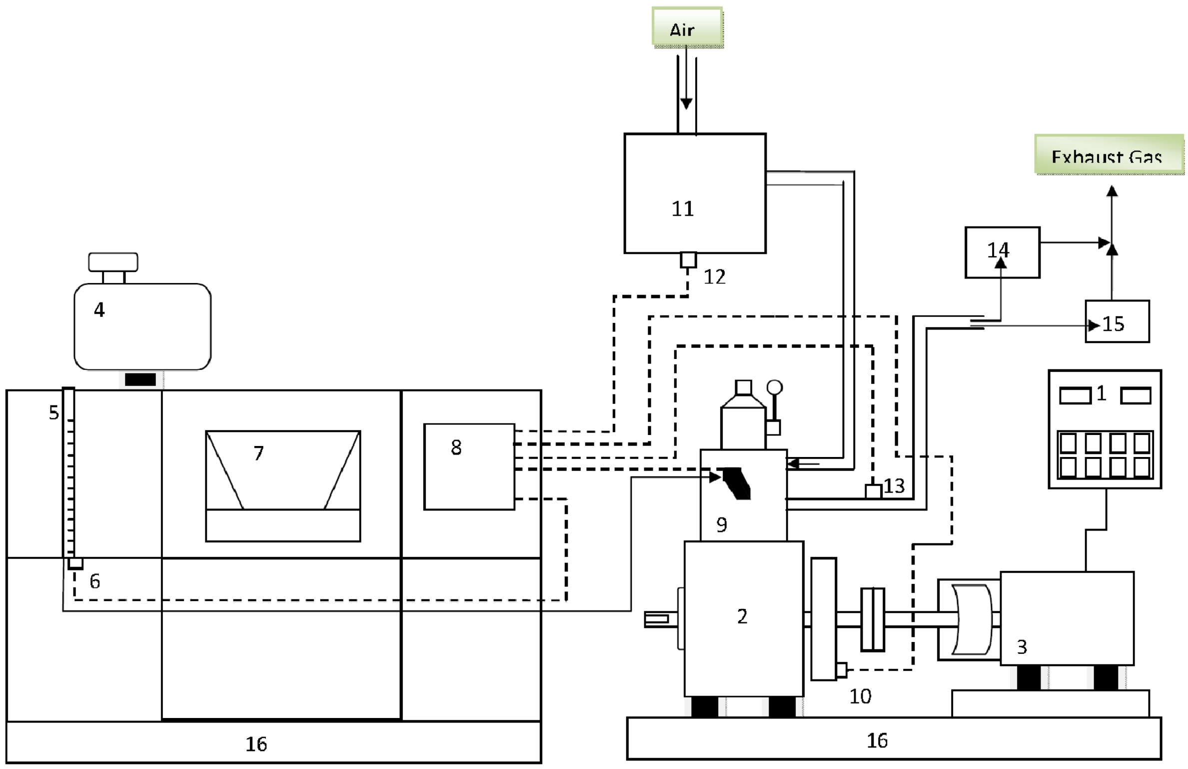

Fig. 3.1 Schematic diagram of experimental setup

Table 3.5 Components of experimental setup

1	Electrical Load	9	Pressure transducer
2	Engine	10	Speed sensor
3	Alternator	11	Air box
4	Fuel tank	12	Air sensor
5	Burette	13	Temperature indicator
6	Fuel sensor	14	Exhaust gas analyzer
7	Computer	15	Smoke meter
8	DAS	16	Bed

Fig. 3.2 Photographic view of engine head during changing compression ratio

By changing the volume of the cylinder head gasket we can change the compression ratio up to a limit.

Fig. 3.3 Photographic view of experimental setup

Table 3.6 Details of instruments

Instrument	Measurement	Range	Accuracy	Uncertainty
Load cell	Loading device	250–6000W	±1W	0.2
Temperature indicator	Exhaust gas measurement	0–900	±1 °C	0.15
Burette	Fuel consumption	1–30 cc	±0.2 cc	1.5
Speed sensor	Speed	0–10,000 rpm	±10 rpm	±1
Exhaust gas analyzer	NO emission	0–5000 ppm	±50 ppm	1
	HC emission	0–20,000 ppm	±10 ppm	0.5
	CO emission	0–10%	0.03%	1
Smoke meter	Smoke density	0–100 %		
Pressure transducer	Cylinder pressure	0–110 bar	±1 bar	0.15
Crank angle encoder	Crank angle		±1	1

An uncertainty analysis was performed using the method described by Holman [68]. The details of instruments used in the study are given in Table 3.6.

The total percentage of the uncertainty of this experiment is calculated as given below.

Total percentage of the uncertainty of this experiment $= [(TFC)^2 + (BP)^2 + (BSFC)^2 + (BTE)^2 + (CO)^2 + (CO_2)^2 + (UBHC)^2 + (NO)^2 + (O_2)^2 + (\text{smoke number})^2 + (EGT)^2 + (\text{uncertainty of pressure pick up})^2 + (\text{CA encoder})^2]^{1/2}$

$= [(1.5)^2 + (0.2)^2 + (1.5)^2 + (1)^2 + (0.03)^2 + (0.5)^2 + (1)^2 + (1)^2 + (1)^2 + (1)^2 + (0.15)^2 + (1)^2]^{1/2}$

$= \pm 3.28\%$

CHAPTER 4

RESULTS AND DISCUSSIONS

Phase I: Preliminary investigation on combustion, performance and emission parameters for DI diesel engine only with optimum injection timing of 20° bTDC and standard nozzle opening pressure of 200 bar.

4.1 Combustion parameters

4.1.1 Pressure crank angle diagram

The variation of combustion pressure with respect to crank angle at full load is shown in Fig. 4.1. Peak pressure mainly depends upon the combustion rate at initial stages, which is influenced by the fuel intake component in the uncontrolled heat release phase. The fuel absorbs amount of heat from the cylinder immediately after injection and the amount of heat results in shorter or longer ignition delay. The peak cylinder pressure decreases at the start of combustion and increases further has been observed from the figure. This may be due to the lower cetane number of UTO, which results in longer ignition delay. The cylinder peak pressure for various compression ratios 16:1, 17:1, 17.5:1 and 18.5:1 occurs 375.5°, 375.9°, 373.2° and 375.1° CA respectively, while in diesel at standard CR 372.4° CA. The cylinder peak pressure of UTO is lower than that of diesel because of higher viscosity and low volatility.

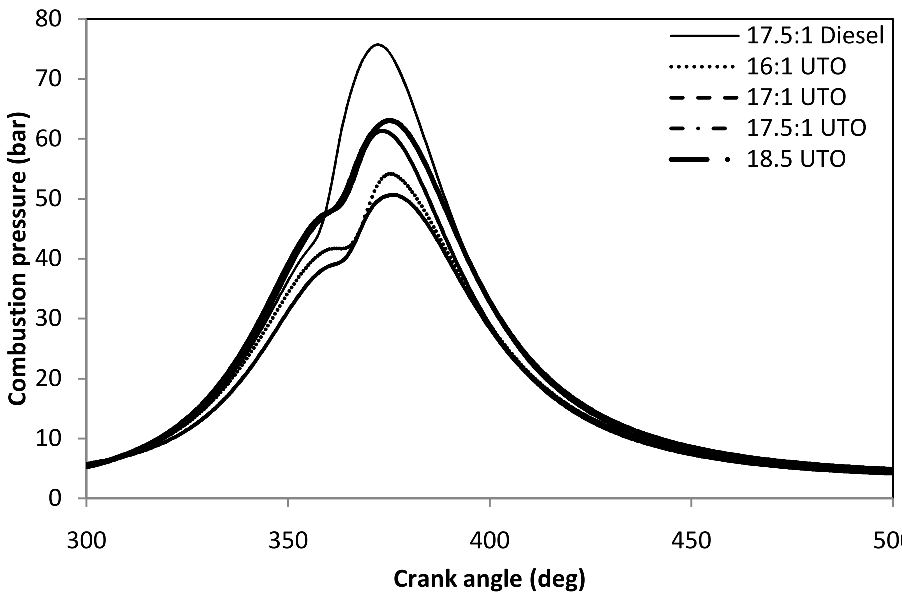

Fig. 4.1 Variation of combustion pressure with crank angle at full load

With compression ratio 17.5:1, the UTO shows cylinder peak pressure 2° CA before than other compression ratio this happens due to faster and complete combustion of fuel inside the combustion chamber [57]. Diesel with standard compression ratio, the cylinder peak pressure occurs at 0.8° CA before compared to UTO at standard CR.

4.1.2 Ignition delay

Ignition delay is the time difference measured in degree crank angle between start of injection and start of ignition of a fuel in diesel engine. The type of fuel is an important parameter affecting the ignition delay. The variation of the ignition delay for the UTO in different CR and diesel with respect to brake power is presented in Fig. 4.2. The ignition delay of the tested compression ratio in this study decreases as the brake power increases. As the load increases the heat prevailing inside the cylinder increases and helps the air fuel mixture to ignite sooner, hence this trend is genuine. It can also be observed from the figure that the ignition delay is found to decrease. Considering the diesel at standard compression ratio as a reference, it is clear that at full load CR 16:1 shows 2.68% longer ignition delay while CR`s 17:1, 17.5:1 and 18.5:1 results in shorter ignition delay of 3.39%, 9.82% and 10.7% with the UTO respectively.

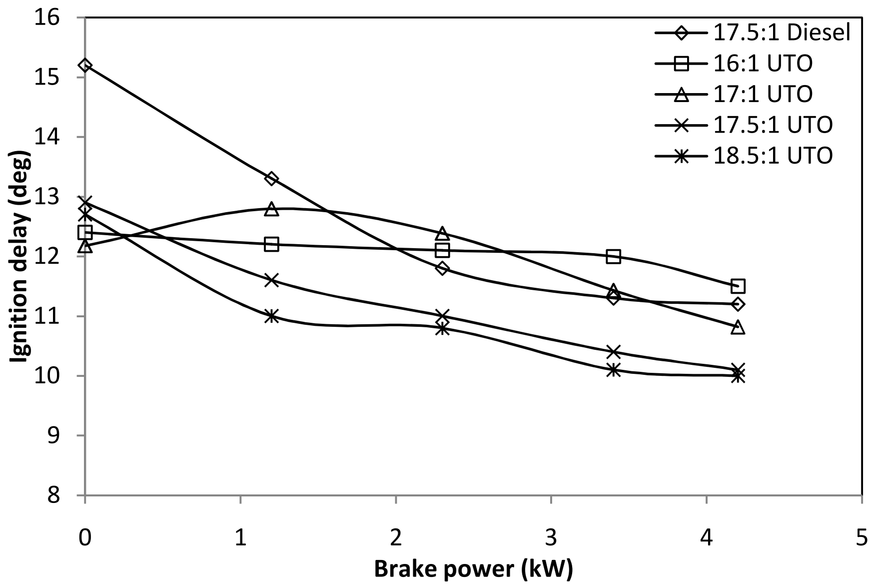

Fig. 4.2 Variation of the ignition delay with brake power

4.1.3 Heat release rate with crank angle

Heat Release Rate (HRR) is the measure of how fast chemical energy of fuel is converted into the thermal energy by combustion. This directly affects rate of pressure rise and accordingly the power produced. The variation in the heat release rate with crank angle is shown in Fig. 4.3 for the UTO at various compression ratios. The heat release is analyzed based on the crank angle variation of the cylinder. It can be observed from the figure that heat release rate increases with lower compression ratio and slightly decrease at higher compression ratio. This may be due to the air entertainment and lower air/fuel mixing rate and effect of viscosity of the UTO.

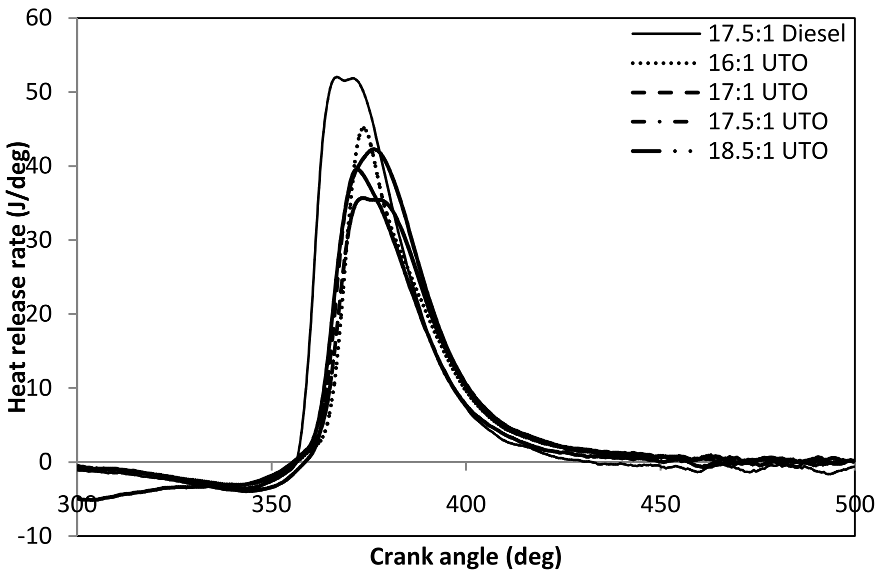

Fig. 4.3 Variation of heat release rate with crank angle at full load

The peak heat release rate at CR 16:1, 17:1, 17.5:1 and 18.5:1 occurs at 373.8°, 373.6°, 372.1° and 375.1° CA respectively. The heat release rate for diesel is found to be higher than that of UTO due to its reduced viscosity and better spray formation [69]. The heat release rate of UTO increases with higher compression ratios which result in reduction in viscosity and good spray formation inside the cylinder.

4.1.4 Max. heat release rate

Figure 4.4 portrays the variation of maximum heat release rate with respect to brake power for the UTO and diesel.

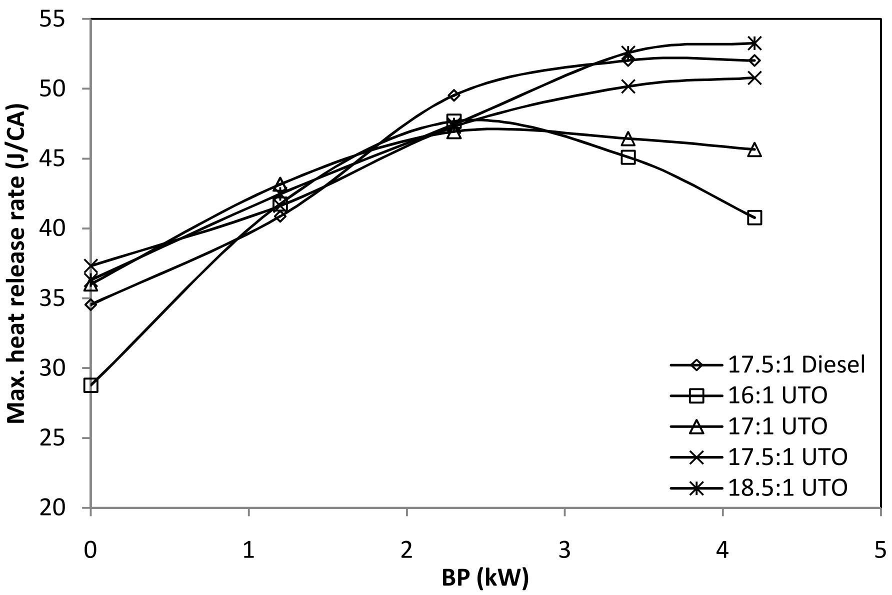

Fig. 4.4 Variation in maximum heat release with brake power

The heat release rate will be less for fuel having high viscosity and less volatility, so that the heat release rate of UTO is always lesser than diesel with lower CR. In comparison with diesel at standard compression ratio, UTO at CR`s 16:1, 17:1 and 17.5:1 shows 21.6%, 12.3% and 2.3% lesser heat release rate, whilw CR 18.5:1 shows higher heat release rate by about 2.4% at maximum brake power. When engine operates in rich mixture, it reaches stoichiometric region at higher compression ratio. More fuel is accumulated in the delay period and it causes rapid heat release.

4.1.5 Peak cylinder pressure

The cylinder peak pressure of a CI engine mainly depends on the amount of fuel accumulated in the delay period and the combustion rate in the initial stages of premixed combustion [70]. Fig. 4.5 shows the variation of the peak cylinder pressure with brake power for the UTO and diesel. Maximum pressure is dependent on compression pressure, time of ignition, atomizing of fuel, and the fuel quality. High compression ratio (pressure) and early ignition gives a higher maximum pressure. In comparison with diesel at standard compression, the UTO at lower CR`s 16:1 and 17:1 shows 21.5% and 9.6% less peak pressure respectively whereas the CR`s of 17.5:1 and 18.5:1 shows 9.8% and 13.9% higher peak pressure than that of diesel.

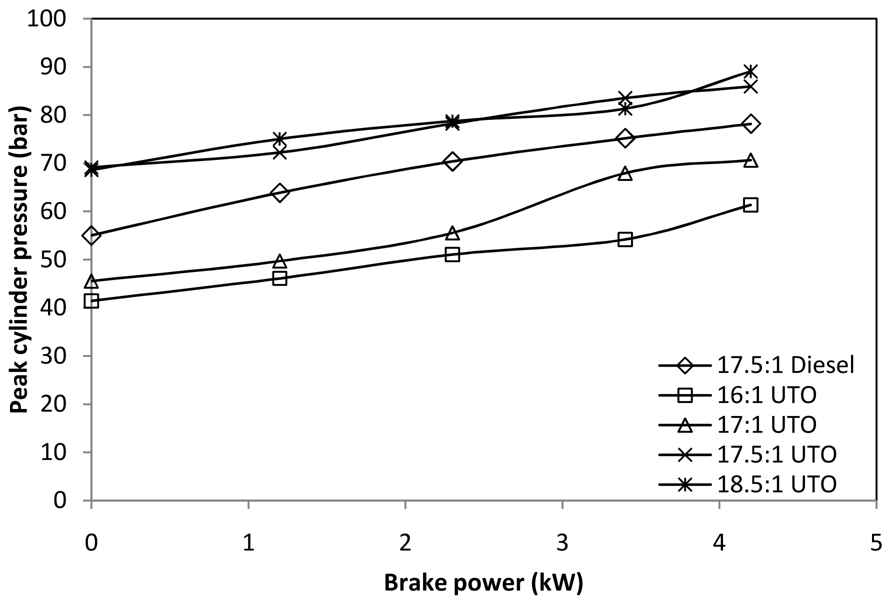

Fig. 4.5 Variation in peak cylinder pressure with brake power

4.1.6 Rate of pressure rise

The rate of pressure rise defines the load that is imposed during the combustion process on the cylinder head and other components. The rate of pressure rise depends on the amount of heat released in the initial stage of combustion and the fuel quality. Higher the rate of pressure rise, higher the load on the piston and other components, which may cause severe damage of the parts. Fig. 4.6 portrays the comparison of rate of pressure rise with respect to the brake power for the UTO and standard diesel.

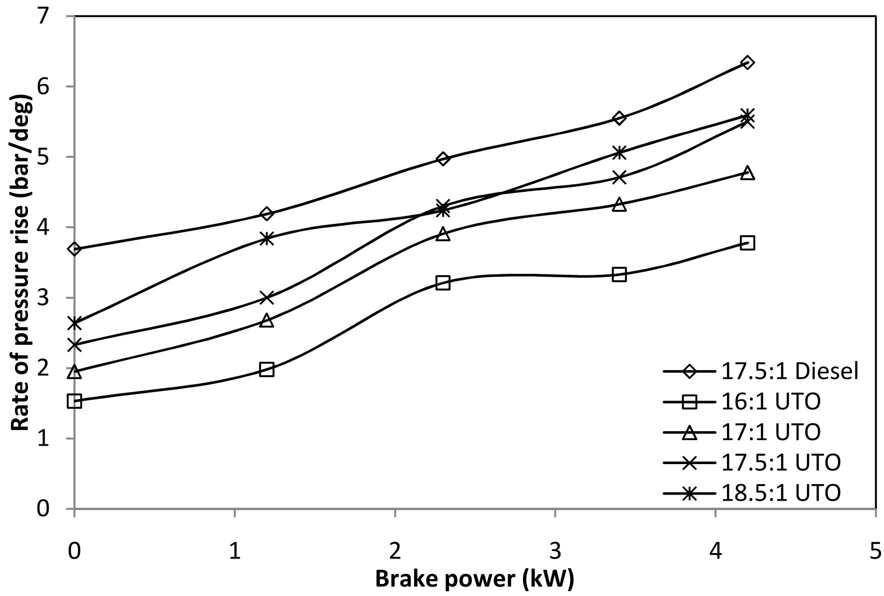

Fig. 4.6 Variation in rate of pressure rise with brake power

4.1.7 Combustion duration

The combustion duration increases as a result of higher fuel consumption that takes part in the combustion. Also the combustion duration increases with increase in engine load or brake power, in general.

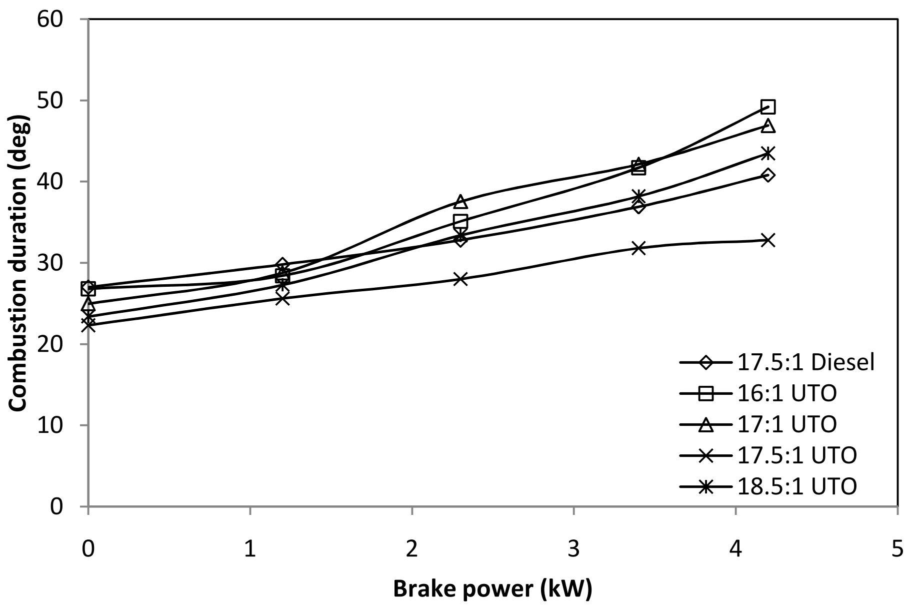

Fig. 4.7 Variation in combustion duration with brake power

Figure 4.7 represents the variation of combustion duration with brake power for the UTO and diesel. The combustion duration of UTO at CR`s 16:1, 17:1 and 18.5 shows the higher combustion duration by 20.5%, 14.9% and 6.6% respectively while the CR 17.5 shows 19.6% less than that of diesel at full load. This reduction in combustion duration is due to the turbulence in the engine cylinder. As the engine speed increases the turbulence inside the cylinder increases, leading to a better heat transfer between burned and unburned zone.

4.2 Performance parameters

4.2.1 Brake thermal efficiency

Brake thermal efficiency give an idea of the output generated by the engine with respect to the heat supplied in the form of fuel. Generally, increasing the compression ratio improved the efficiency of the engine. This improvement in performance of the engine at higher CR is due to the reduced ignition delay. Fig. 4.8 shows the variation of the BTE with brake power for the UTO and standard diesel.

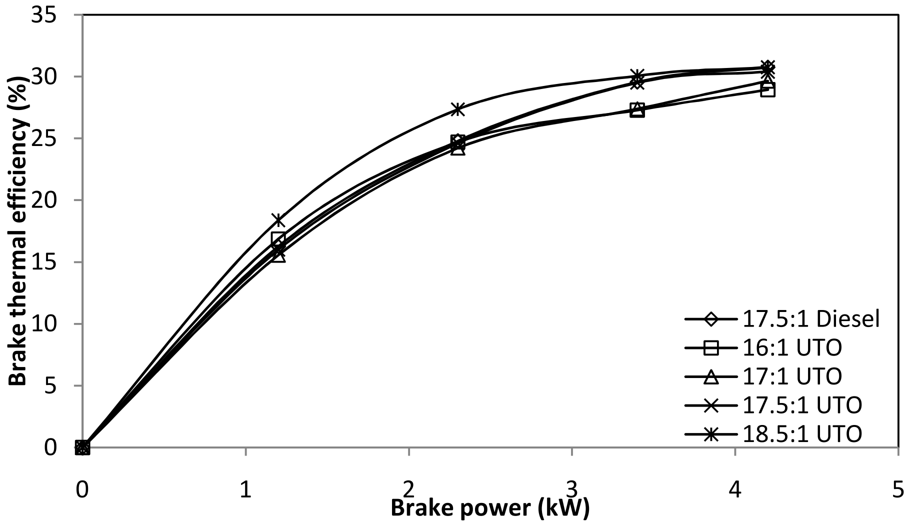

Fig. 4.8 Variation in brake thermal efficiency with brake power

The values of the BTE for diesel and the UTO are found to be about 30.7% and 30.4% at standard CR and maximum brake power. The brake thermal efficiency of UTO is 28.9%, 29.6% and 30.7% at compression ratio of 16:1, 17:1 and 18.5:1. With increase in CR results in improvement of the BTE [71]. This can be attributed to better combustion and better intermixing of air and fuel inside the combustion chamber.

4.2.2 Brake specific energy consumption

An important parameter to measure the engine performance is the specific energy consumption. It is the product of brake specific fuel consumption and lower heating value.

Fig. 4.9 shows the variation between brake specific energy consumption and brake power. As the fact, the BSEC decreases with increase in engine load.

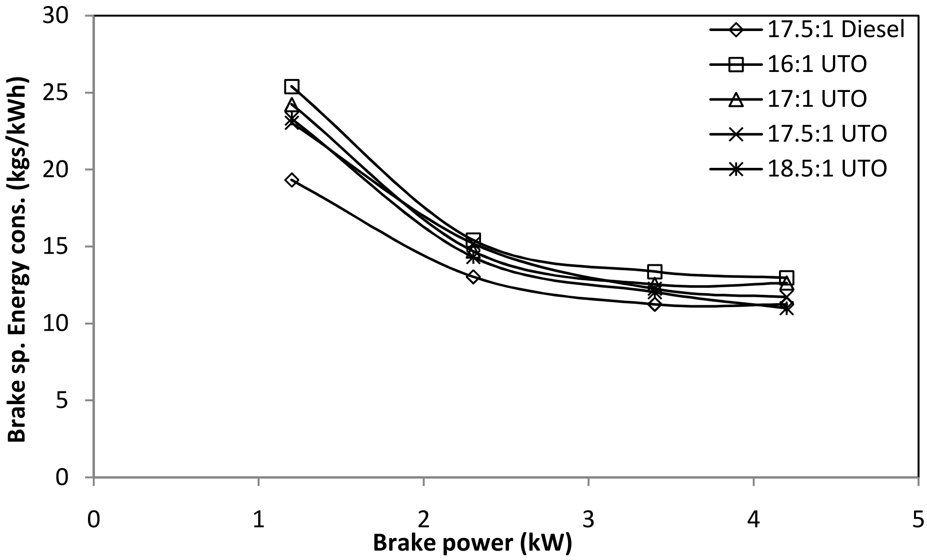

Fig. 4.9 Variation in brake specific energy consumption with brake power

Compare to diesel engine at standard situation, UTO shows 15.5%, 12.4% and 4.3% higher consumption at CR`s 16:1, 17:1 and 18.5:1 while CR of 18.5:1 consumes 2.1% lower than diesel. The increase in the fuel consumption is due to fuel density, viscosity and heating value, but with higher compression ratio lesser value of SEC is apparently desirable.

4.2.3 Exhaust gas temperature

During the combustion inside cylinder the temperature is very high, but with expansion there is a great reduction in the exhaust gas temperature. So the exhaust gas temperature is strongly dependent on in-cylinder temperature and expansion process.

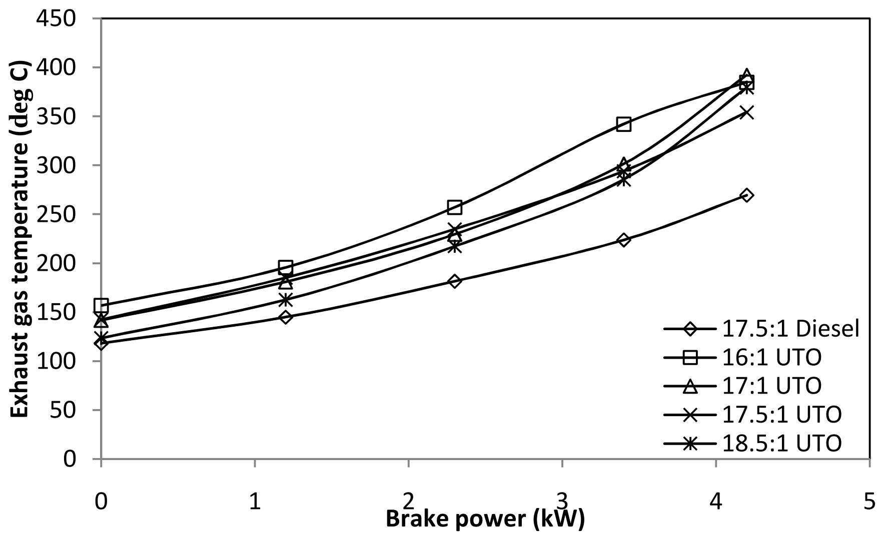

Fig. 4.10 Variation in exhaust gas temperature with brake power

Figure 4.10 portrays the variation of the exhaust gas temperature with brake power for diesel and UTO. The exhaust gas temperature increases with an increase in the engine load for the UTO and diesel [71]. It is higher for the UTO than that of diesel in the entire operation as a result of higher viscosity and density of UTO. The exhaust gas temperature of UTO at compression ratio of 16:1, 17:1, 17.5:1 and 18.5:1 are 385^o, 392^o, 354^o and 350^{o}C respectively whereas in diesel 269^{o}C. The reason for the reduction in the exhaust gas temperature at increased CR is due to improved energy conversion of UTO as compared to that of diesel [72].

4.3 Emission parameters

4.3.1 Carbon monoxide (CO) emission

The amount of CO increases due to less availability of air , poor mixing of air with fuel and rise in temperature in the combustion chamber. A small amount of CO is also occurs due to fuel viscosity and fuel spray quality. Fig. 4.11 portrays the percentage variation of the carbon monoxide emission with brake power for UTO compared to diesel at different compression ratio.

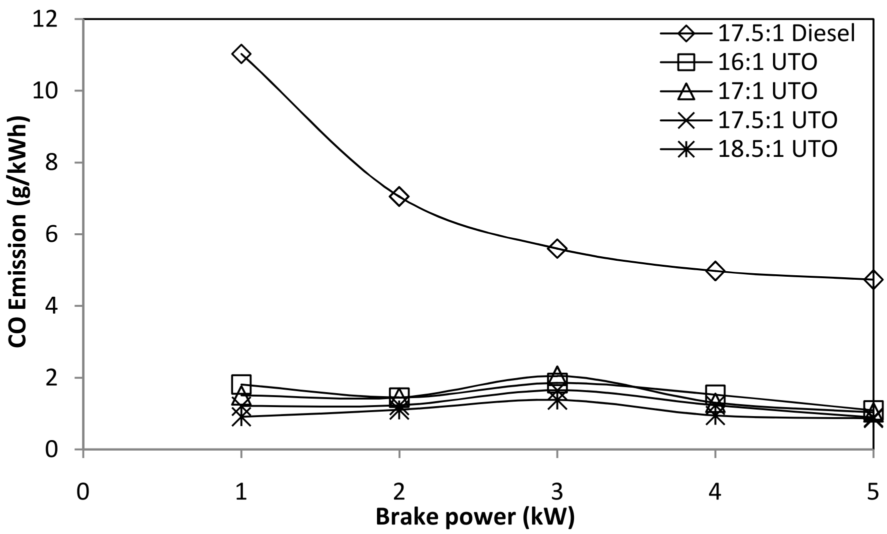

Fig. 4.11 Variation in CO emission with brake power

It can be observed from the figure that the CO emission is higher for the UTO compared to that of diesel at maximum brake power for all compression ratios. Lower CRs of 16:1, 17:1 shows 20%, 18% higher CO than UTO operation at standard compression ratio, whereas 18.5:1 CRs shows20% less CO emission at maximum brake power. The decrease in the CO emission may be due to better combustion and oxygen enrichment of the fuel [53].

4.3.2 Carbon di-oxide (CO_2) emission

More amount of CO_2 indicates the complete combustion of fuel in the combustion chamber and it is also related to the exhaust gas temperature. The excess amount of CO_2 in the atmosphere leads to global warming and environmental problems. These emitted CO_2 are absorbed by the plants to maintain constant percentage in atmosphere. CO_2 is always less in the UTO fuel compared with diesel as shown in Fig. 4.12.

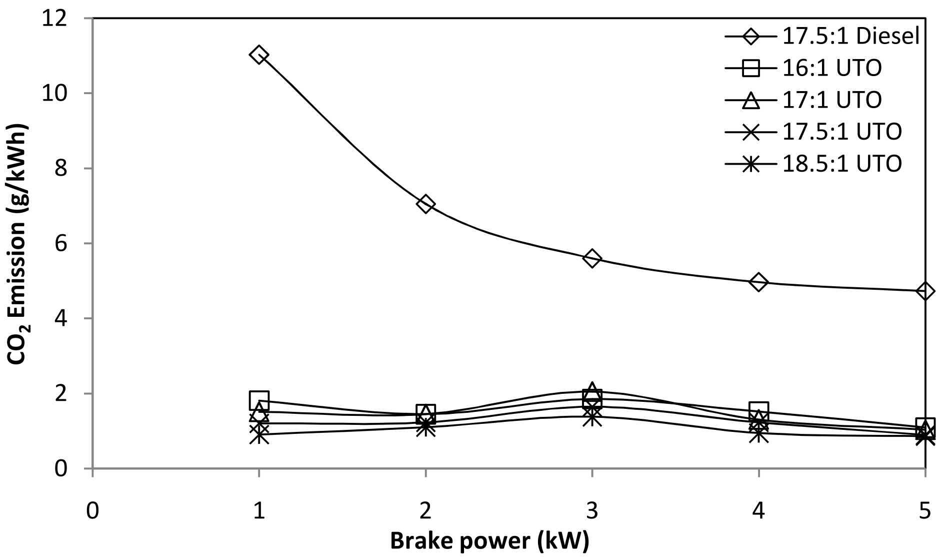

Fig. 4.12 Variation in CO_2 emission with respect to brake power

Compared to standard CR of the UTO, CRs 16:1 and 17:1 shows 2.6% and 1.3% higher whereas 18.5:1 shows lesser CO_2 emission by 1.3% at full load. At CR 18.5 the CO_2 emission is lesser due to incomplete combustion and lack of oxygen.

4.3.3 Hydrocarbon (HC) emission

The reason for the HC emission in a CI engine are wall deposit absorption, oil film absorption, crevice volume, incomplete combustion etc. During the combustion process HC particles condenses onto the surface of solid carbon soot. Most of this burned and only a small percentage of carbon soot comes out from the cylinder which contributes to the HC emission of the engine [3]. Fig. 4.13 portrays the percentage variation of the HC emission with brake power for UTO compared to diesel at different compression ratio. The HC emission in UTO is much higher than that of diesel for all CRs. It can be observed from the figure that the HC emission is higher by about 36.6% and 12.7 % for CRs of 16:1 and 17:1 compared to standard CR of UTO while the CR of 18.5:1 shows 30.6% lower at maximum brake power due to delays in ignition, which results to insufficient heat of compression so HC emissions decreases [56]. Fuel viscosity and spray quality is also responsible for increase in HC with UTO.

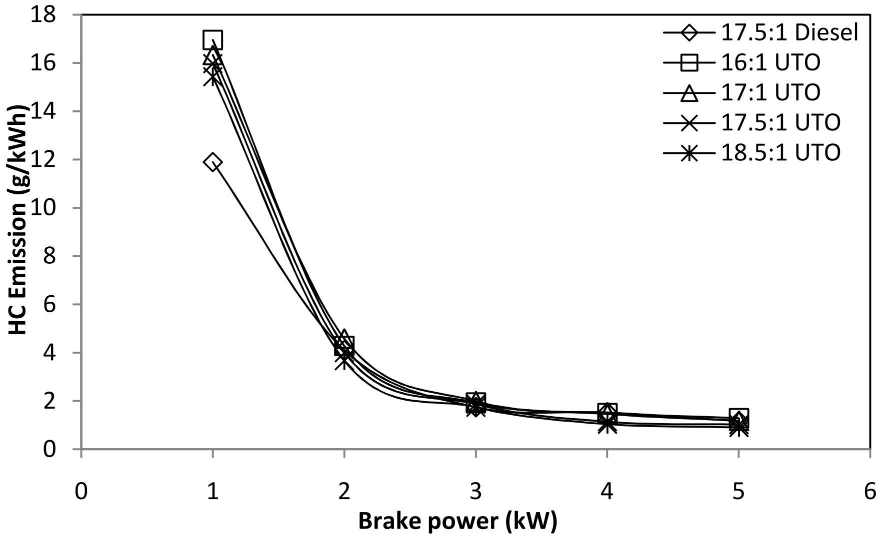

Fig. 4.13 Variation in HC emission with brake power

4.3.4 Nitrogen oxide (NO) emission

An engine can have up to 2000 ppm of oxides of nitrogen in the exhaust gas. With higher compression ratio, the cylinder pressure and high temperature contribute to dissociate diatomic N into monatomic N, thus resulting in more NO formation. The reduction in the NO emission is the prime objective of the engine researcher.

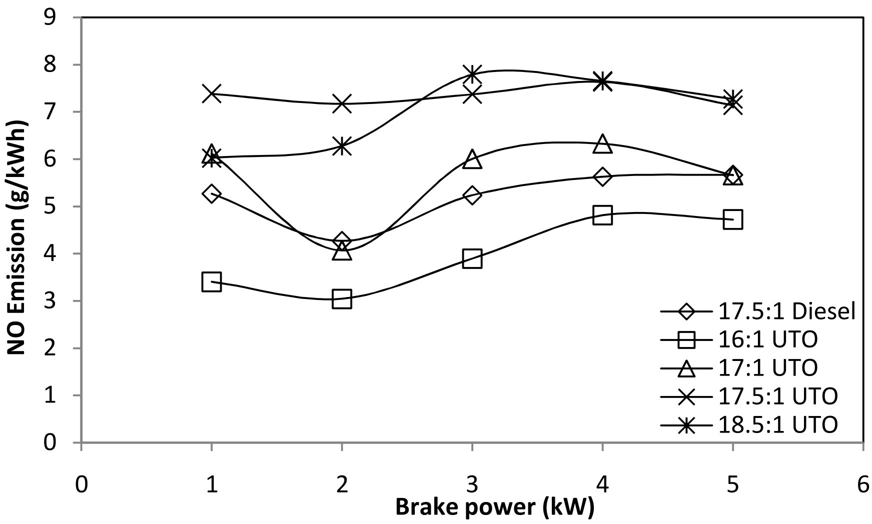

Fig. 4.14 Variation in NO emission with brake power

Figure 4.14 portrays the percentage variation of the NO emission with brake power for UTO compared to diesel at different compression ratio. The NO emission in a CI engine strongly depends on the combustion temperature and oxygen availability. The NO emission for the UTO with CRs of 16:1 shows lower by 6.2% whereas 17:1, 17.5:1 and 18.5:1 are found to be higher by 12.3%, 27.5% and 30.6% to that of diesel at maximum brake power. In comparison with standard CR of UTO, when lower the CR to 16:1 and 17:1 the NO emission is found to be lower by 33.7%. and 15.2% and with higher CR of 18.5:1 higher by 3.1% at maximum brake power. With higher CR, the NO emission for the UTO is increased due to high in-cylinder (peak) temperature [51].

4.3.5 Smoke opacity

Smoke is higher when a fuel's ratio of hydrogen to carbon is less than two [71]. Fig. 4.15 portrays the percentage variation of the smoke opacity with brake power for the UTO compared to diesel at different compression ratios.

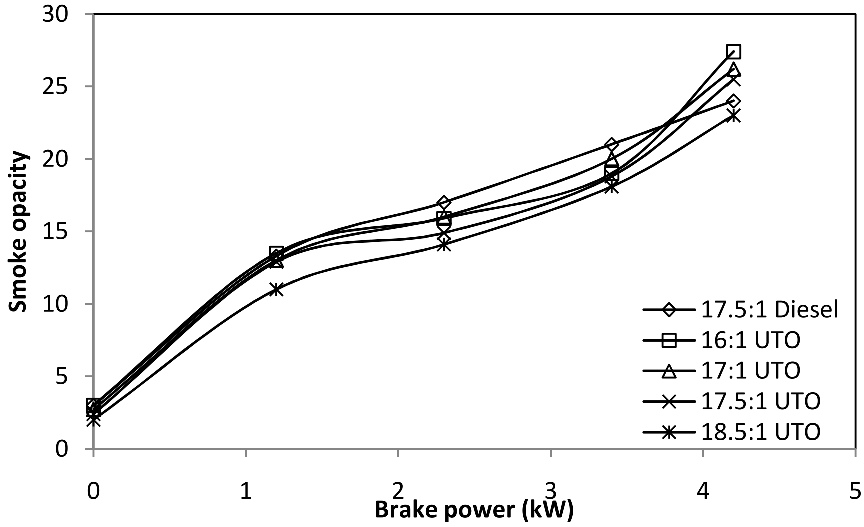

Fig. 4.15 Variation in smoke opacity with brake power

The hydrogen to carbon ratio of the UTO is lower by about 49% than that of diesel. Among all the compression ratios, UTO with CR 18.5 exhibits the lowest smoke opacity by 4.1% compared to diesel. At maximum brake power, the CR of 16:1 and 17.1 UTO shows 14.1% and 9.2% increase while higher CR 18.5:1 shows 8.6% less smoke opacity compared to UTO at standard compression ratio. This may be due to the maximum temperature during the combustion increases and this, in turn, decreases smoke opacity [72].

Phase II: Combustion, performance and emission parameters for DI diesel engine operated at optimum injection timing of 20° bTDC and optimum nozzle opening pressure of 230 bar.

4.4 Combustion parameters

4.4.1 Pressure crank angle diagram

The variation of the cylinder pressure with respect to crank angle for different CRs for the UTO are compared with the reference data and shown in Fig. 4.16. The maximum cylinder pressure for diesel is 75 bar at 372^{o}CA. The occurrence of the ignition timing of UTO with CR 17:1, 17.5:1 and 18.5:1 is at 1, 1.5, 2.5^{o}CA respectively earlier than that of diesel operation at full load. The CR 16:1 shows approximately 1^{o}CA later occurrence of ignition than diesel at full load. Increasing the compression ratio reduces the ignition delay, but results in increase of maximum cylinder pressure as a result of higher heat released. The maximum pressure occurs at high compression ratio due to rapid and complete combustion of fuel inside the combustion chamber [57]. This can be evidenced from Fig. 4.16.

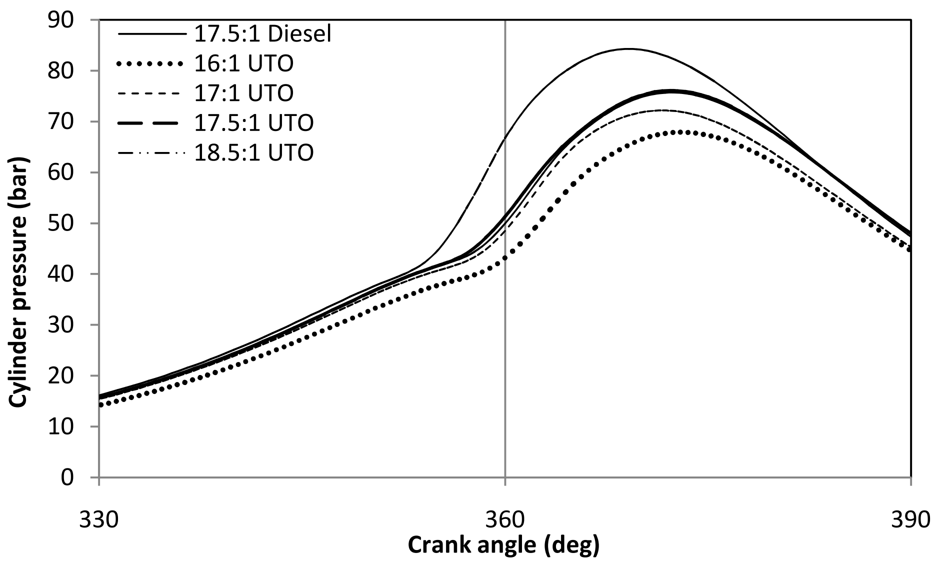

Fig. 4.16 Variation of combustion pressure with crank angle at full load

4.4.2 Ignition delay

Ignition delay is the time difference measured in degree crank angle between start of injection and start of ignition of a fuel in diesel engine. The type of fuel is an important parameter affecting the ignition delay. The variation of ignition delay with brake power is shown in Fig. 4.17. Higher CR of 18:1 shows a shorter ignition delay while the lower CR of 17:1 and 16:1 shows a longer ignition delay than that of UTO at maximum brake power. Longer mixing of fuel with air gives rise to larger premixed combustion phases and conversely less diffusion combustion occurs which results in increased ignition delay at 16:1 CR. Increasing the CR also results in an increased gas temperature. As a result, the ignition delay tends to decrease with increase in the CR. The value of the ignition delay at full load for diesel and UTO at standard CR is 11.2 and 9.8^o CA respectively. At the CR's 16, 17 and 18.5:1 the value of ignition delay are 10, 9.9 and 9.5^o CA respectively at full load.

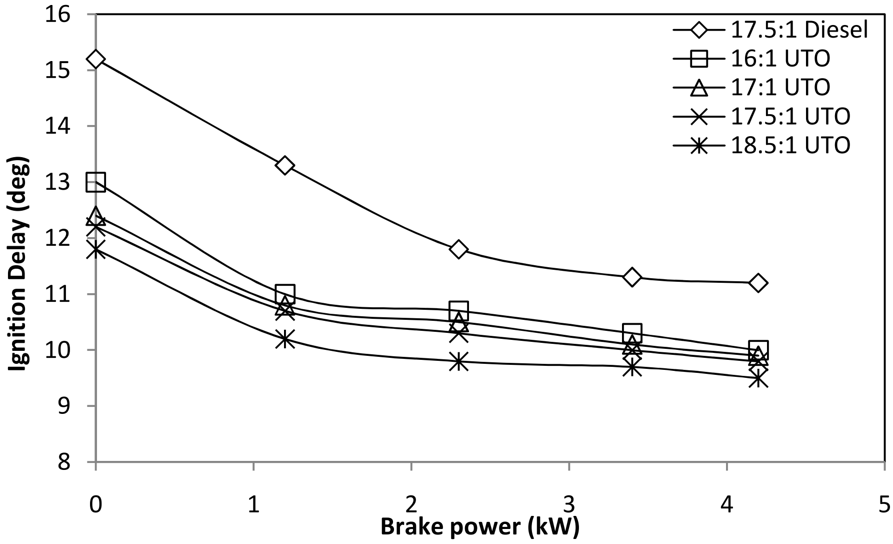

Fig. 4.17 Variation of ignition delay with brake power

4.4.3 Heat release rate with crank angle

Heat Release Rate (HRR) is the measure of how fast chemical energy of fuel is converted into the thermal energy by combustion. The heat release rate is analyzed based on the changes in crank angle variation.

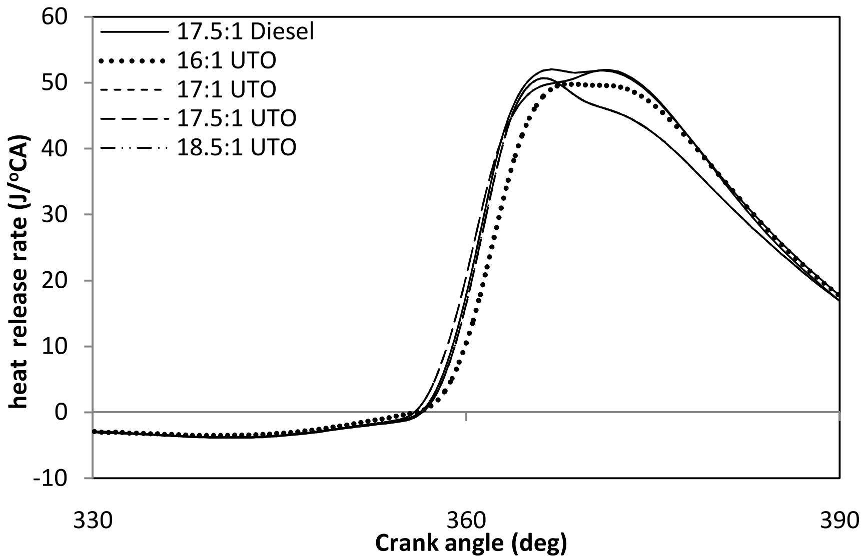

Fig. 4.18 Variation of heat release rate with crank angle at full load

This directly affects the rate of pressure rise and accordingly the power produced. The variation of the heat release rate with crank angle is shown in Fig. 4.18. It can be observed from the figure that the diesel is the highest followed by CR 16, 17, 17.5 and 18.5:1 for the UTO. The heat release rate of diesel is higher because of less viscosity and better spray formation [69]. Higher compression ratio, increases the cylinder pressure and temperature more. The heat release rate values are found to be 52, 49.7, 50.6, 51.9 and 52.9 J/^{o}CA for diesel, UTO with CR 16, 17, 17.5 and 18.5:1 respectively at full load.

4.4.4 Maximum heat release rate

Fig. 4.19 portrays the variation of maximum heat release rate with respect to brake power for UTO and diesel.

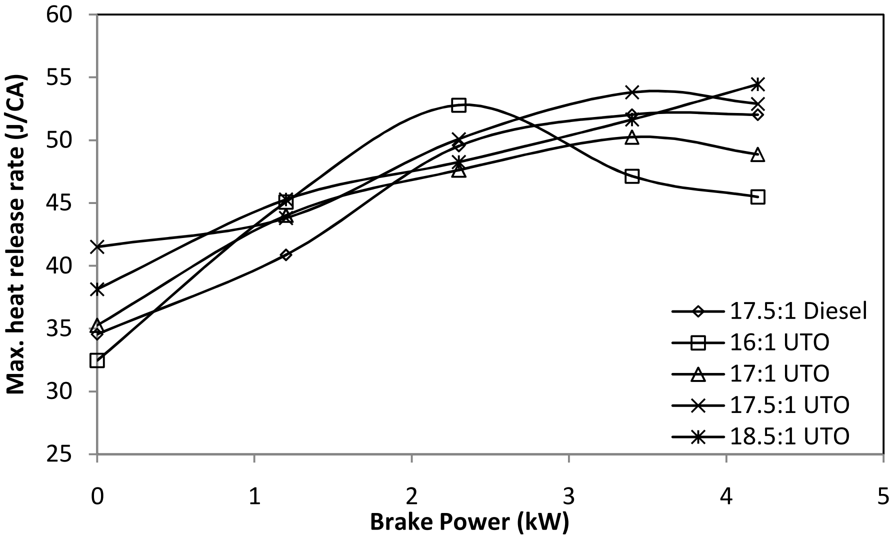

Fig. 4.19 Variation in maximum heat release rate with brake power

The heat release rate will be less for fuel having high viscosity and less volatility, so that the heat release rate of UTO is always lesser than diesel. In comparison with diesel at standard compression ratio, the UTO at CR`s 16:1 and 17:1 shows 12.5% and 6.0% lesser heat release rate, while 17.5:1 and 18.5:1 shows 1.6% and 6.7% lesser heat release rate at maximum brake power. When the engine operates in rich mixture, it reaches stoichiometric region at higher compression ratio. More fuel is accumulated in the combustion phase and it causes rapid heat release during uncontrolled combustion phase [70].

4.4.5 Peak cylinder pressure

The cylinder peak pressure of a CI engine mainly depends on the amount of fuel accumulated in the delay period and the combustion rate in the initial stages of premixed combustion [70]. Peak cylinder pressure decreases at the beginning of the combustion and then further increases. Fig. 4.20 shows the variation of the peak cylinder pressure with brake power for the UTO and diesel. Maximum pressure is dependent on compression pressure, time of ignition, atomizing of fuel, and the fuel quality. High compression ratio (pressure) and early ignition gives high maximum pressure.

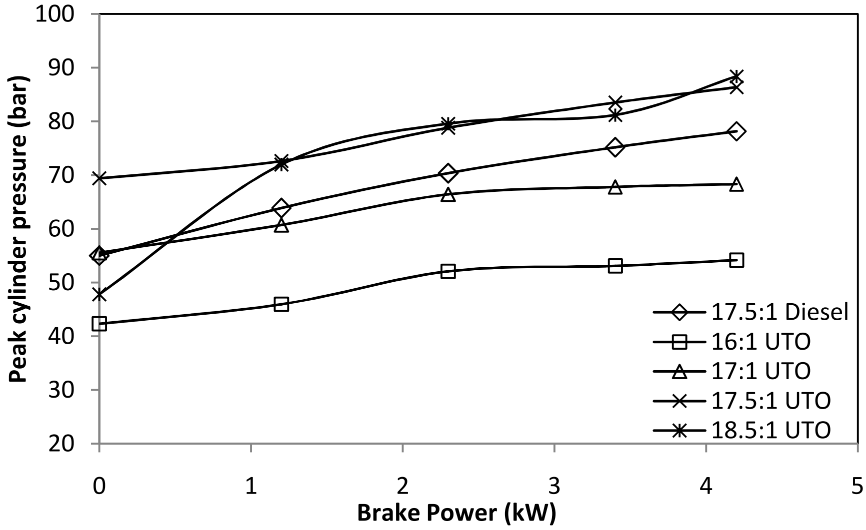

Fig. 4.20 Variation in peak cylinder pressure with brake power

In comparison with diesel at standard compression, the UTO at lower CR`s 16:1 and 17:1 shows 30.7% and 12.5% less peak pressure respectively whereas the CR`s of 17.5:1 and 18.5:1 shows 10.5% and 13.1% higher peak pressure than that of diesel.

4.4.6 Rate of pressure rise

Figure 4.21 portrays the comparison of rate of pressure rise with respect to the brake power for the UTO and standard diesel.

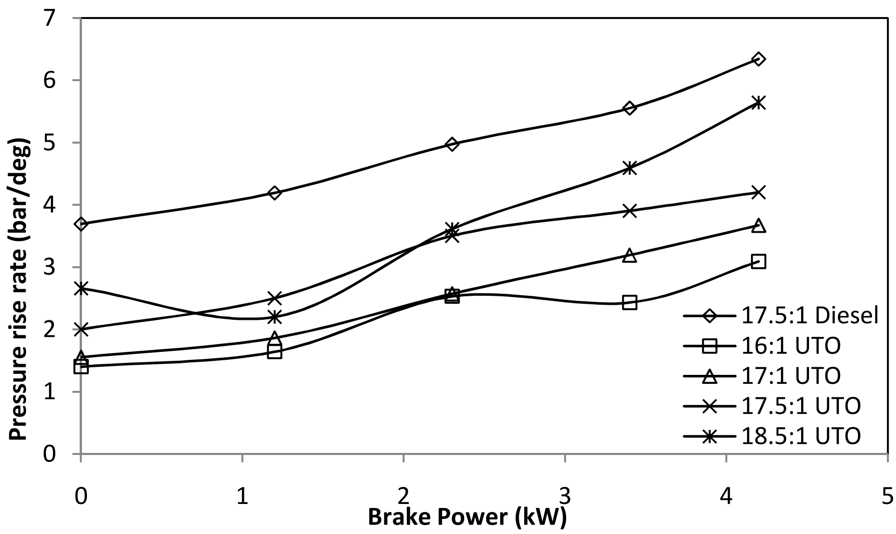

Fig. 4.21 Variation in rate of pressure rise with brake power

The rate of pressure rise defines the load that is imposed during the combustion process on the cylinder head and other components. The rate of depends on the amount of heat released in the initial stage of combustion and the fuel quality. Higher the rate of pressure rise, higher the load on the piston and other components, which may cause severe damage of the parts [44]. The rate of pressure rise is highest for diesel compared to UTO at all CR`s because of longer ignition delay and shorter combustion duration of diesel. At CR's of 16:1, 17:1, 17.5:1 and 18.5:1 UTO shows 51.2%, 42.1%, 33.7% and 11.0% respectively lesser pressure rise at full load than that of diesel. In comparison with standard CR of 17.5, higher CR of 18.5 shows 22.6% higher rate of pressure rise for the UTO.

4.4.7 Combustion duration

The start of heat release to the end of heat release is accounted in the combustion duration. The combustion duration increases as a result of higher fuel consumption that takes part in the combustion. Also the combustion duration increases with increase in engine load or brake power, in general. Fig. 4.22 represents the variation of combustion duration with brake power for the UTO and diesel.

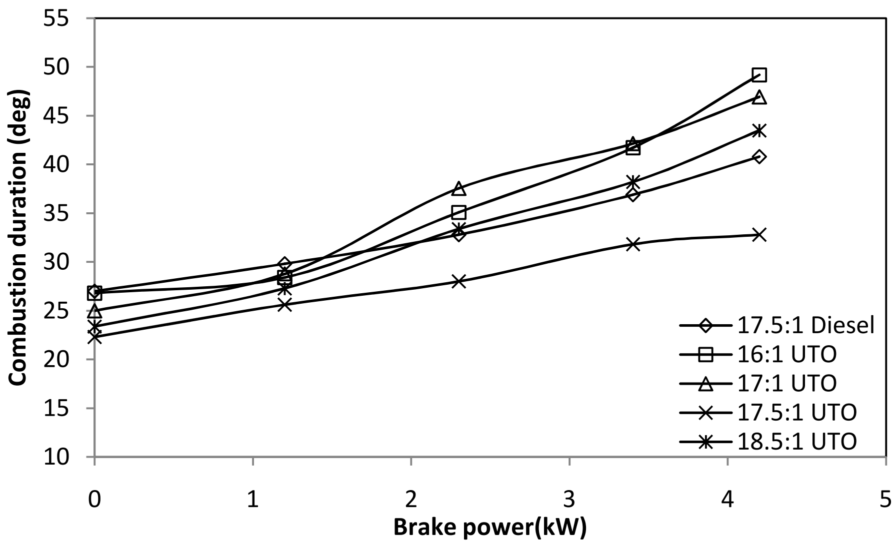

Fig. 4.22 Variation in combustion duration with brake power

The combustion duration of UTO at CR`s 16:1, 17:1 and 18.5 shows the higher combustion duration by 20.5%, 14.9% and 6.6% respectively while the CR 17.5 shows 19.6% less than that of diesel at full load. This reduction in combustion duration is due to the turbulence in the engine cylinder and because of oxygen present in the UTO. As the engine speed increases the turbulence inside the cylinder increases, leading to a better heat transfer between burned and unburned zone.

4.5 Performance parameters

4.5.1 Brake thermal efficiency

Generally, increase in compression ratio improves the efficiency of the engine because of reduced ignition delay. Fig. 4.23 illustrates the variation of the brake thermal efficiency with brake power for the UTO with different compression ratio and diesel.

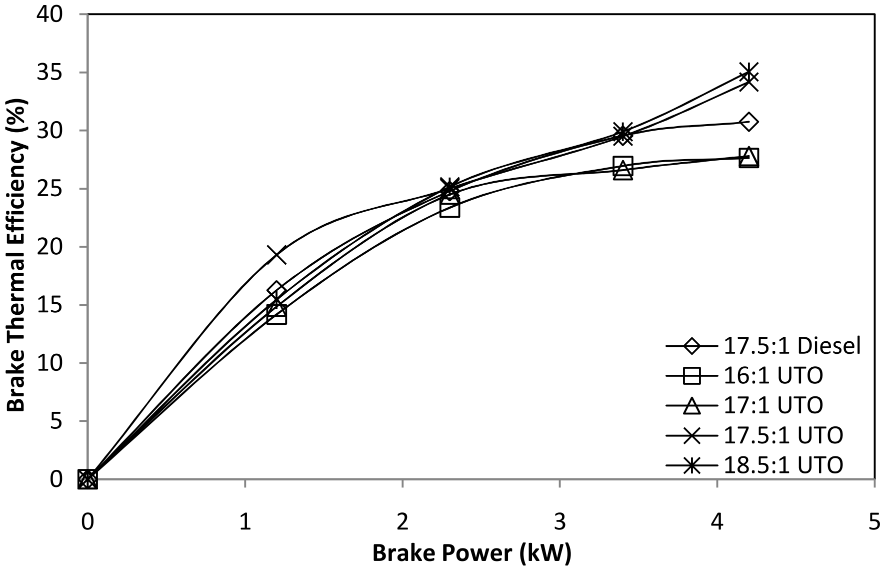

Fig. 4.23 Variation of the brake thermal efficiency with brake power

The values of brake thermal efficiency for diesel and the UTO are 28.6 and 34.1% respectively with standard compression ratio at maximum brake power. The brake thermal efficiency of UTO is 27.6, 27.8 and 35% at compression ratio of 16:1, 17:1 and 18.5:1 respectively. This can be attributed to the better combustion and better intermixing of the fuel and air [60]. The increase in the brake thermal efficiency also seen with increase in load due to lesser losses, due to low calorific value of the UTO [69]. As the fact, the UTO is volatile than the diesel but at higher CR of 18.5 shows relatively more improvement because of higher temperature.

4.5.2 Brake specific energy consumption

An important parameter to measure the engine performance is the specific energy consumption. It is simply described as the product of brake specific fuel consumption (BSFC) and lower heating value (LHV). Fig. 4.24 shows the variation between brake specific fuel consumption and brake power. As the fact, the BSEC decreases with increase in engine load. Compare to diesel engine at standard situation, UTO shows 18.5%, 11.9% and 0.5% higher consumption at CR`s 16:1, 17:1 and 18.5:1 while CR of 18.5:1 consumes 6.7% lower than diesel. The higher CR of 18.5 shows 7.2% lesser BSEC than the standard CR of 17.5 with the UTO. The increase in the fuel consumption is due to fuel density, viscosity and heating value, but with higher compression ratio lesser value of BSEC is apparently desirable because of better atomization which is associated with a marginal delay in admission due to high needle lift pressure during same period, hence less fuel goes inside the combustion chamber [70].

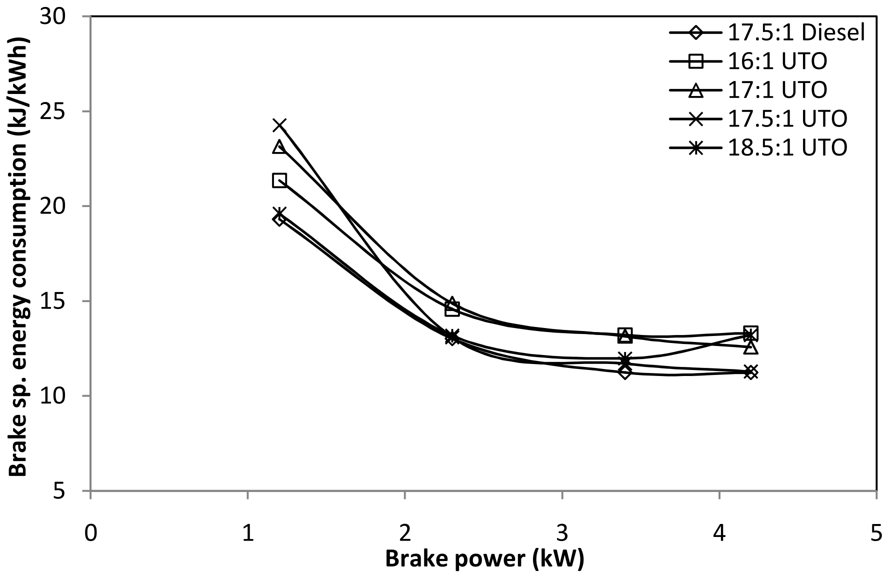

Fig. 4.24 Variation in brake specific energy consumption with brake power

4.5.3 Exhaust gas temperature

Figure 4.25 portrays the variation of the exhaust gas temperature with brake power for diesel and the UTO.

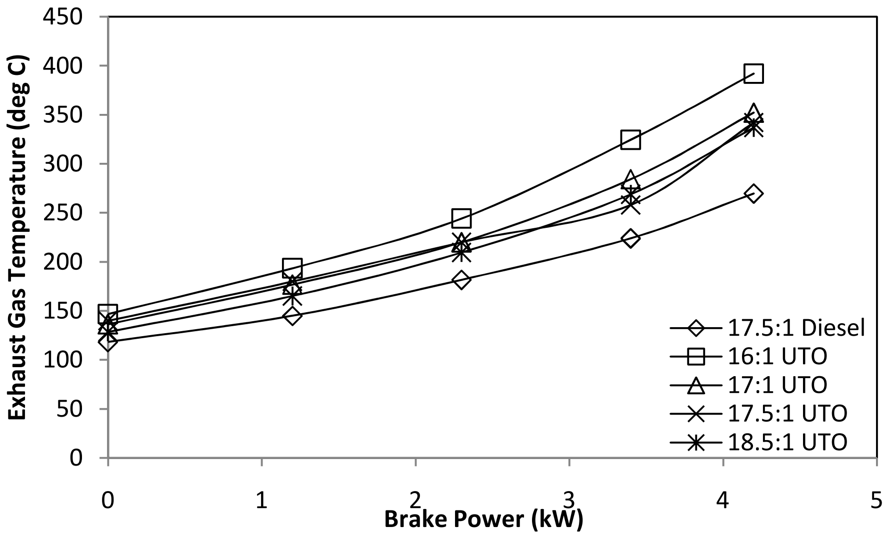

Fig. 4.25 Variation of the exhaust gas temperature with brake power

The exhaust gas temperature increases with an increase in the engine load for the UTO and diesel as expected [71]. It is higher for the UTO than that of diesel in the entire operation as a result of higher viscosity and density of the UTO. The exhaust gas temperature of diesel and the UTO with the CR 16, 17, 17.5 and 18.5:1 are 269, 392, 352, 342 and 336^{o}C respectively. The reduction in the exhaust gas temperature at increased CR may be due to improved energy conversion and lower heating value of the UTO as compared to that of diesel [72]. Another possible reason for the reduction in exhaust gas temperature at higher CR is increase in air temperature inside the cylinder which reduces the ignition lag due to better and more complete burning of the fuel.

4.6 Emission Parameters

4.6.1 Carbon monoxide (CO) emission

Carbon monoxide is an odorless and colorless gas but poisonous in nature. When engine is operated with the rich fuel-air equivalence ratio, it is generated. The CO emission is caused due to less availability of air, poor mixing of air with fuel and rising temperature in the combustion chamber. The variation of the carbon monoxide emission for the UTO and diesel for different engine loads is shown in Fig. 4.26. It shows that the reduction of CO emission by about 75% with the UTO at CR 18.5 than that of the diesel at 17.5:1 CR. It can be observed from the figure that the CO emission is higher for the UTO at 17.5:1 CR compared to that of diesel at full load.

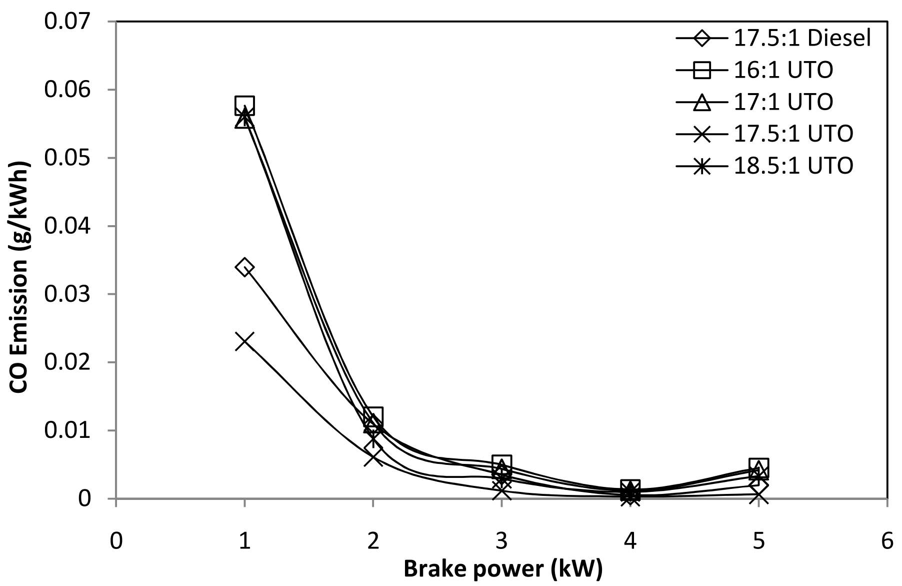

Fig. 4.26 Variation of carbon monoxide with brake power

Higher compression ratio shows a lower CO emission due to better combustion and oxygen enrichment of the fuel [53]. The heat generated is more inside the cylinder due to higher compression. As a result, evaporation rate is increased and results in better fuel air mixing. Thus, the CO emission reduced with increased CR.

4.6.2 Carbon dioxide (CO_2) emission

More amount of CO_2 indicates the complete combustion of fuel in the combustion chamber and also relates to the exhaust gas temperature. The excess amount of CO_2 in the atmosphere leads to global warming and environmental problems. These emitted CO_2 are absorbed by the plants to maintain constant percentage in atmosphere. CO_2 is always less in UTO fuel compared with diesel as shown in Fig. 4.27.

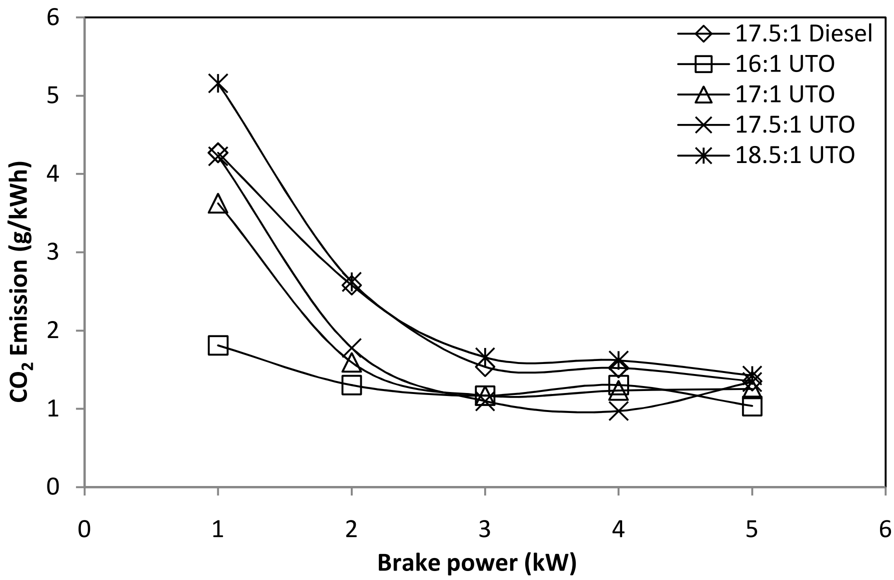

Fig. 4.27 Variation in CO_2 emission with respect to brake power

Compared to standard CR 17.5 of UTO, CRs 16:1 and 17:1 shows 2.6% and 1.3% higher whereas 18.5:1 shows lesser CO_2 emission by 1.3% at full load. At CR 18.5 the CO_2 emission is lesser due incomplete combustion and insufficient supply of oxygen.

4.6.3 Hydrocarbon (HC) emission

Hydrocarbon emission is caused by the longer ignition delay and accumulation of fuel in the combustion chamber. The variation of the unburned hydrocarbon emission for UTO and diesel for different brake power is shown in Fig. 4.28. It can be observed from the figure that the HC emission is higher by about 11.7% for compression ratio of 16:1 compared to that of diesel at maximum brake power, whereas for 17:1, 17.5:1 and 18.5:1 shows 22, 74 and 54% lower HC emission. As the compression ratio increases it shows a lower HC emission, and lower CR shows higher HC emission due to longer ignition delay. More fuel is accumulated in the delay period and as a result higher HC emission is formed with lower CR than standard CR for UTO [56].

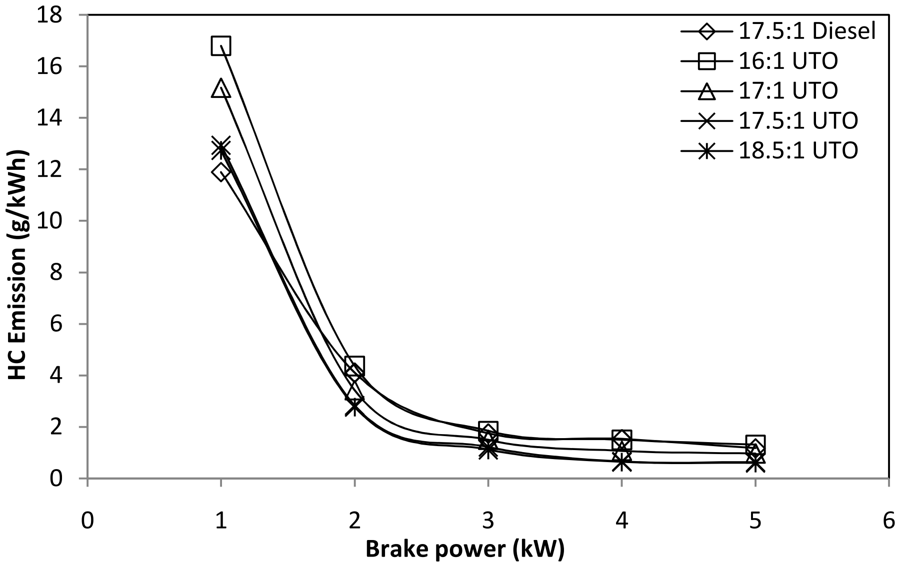

Fig. 4.28 Variation of unburnt hydrocarbon with brake power

4.6.4 Nitric oxide (NO) emission

An engine can have up to 2000 ppm of oxides of nitrogen in the exhaust gas [3]. The variation of the NO emission for UTO with different CR and diesel for different engine loads is shown in Fig. 4.29.

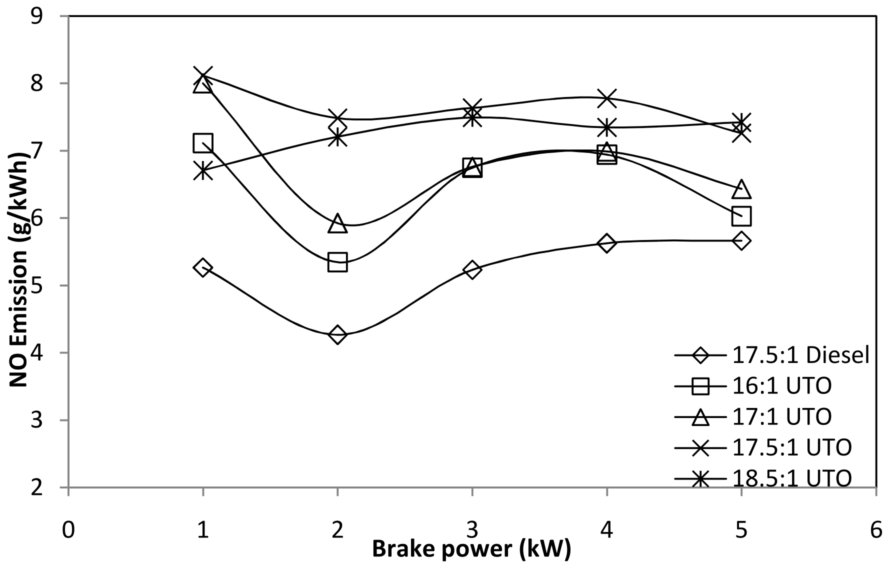

Fig. 4.29 Variation of the NO emission with brake power

The NO emission in a CI engine strongly depends on the combustion temperature and the oxygen availability. At lower compression ratio nitrogen exists as diatomic molecule.

With higher CR there is reduction in ignition delay the cylinder temperature will be more hence the NO emission will be more. The NO emission for diesel and UTO at CR 17.5:1 is 5.6 and 7.2 g/kWh respectively at maximum brake power. The NO emissions for UTO at CR 16, 17 and 18.5:1 are 6, 6.4 and 7.4 g/kWh at maximum brake power. Higher CR shows higher NO emission than that of diesel at maximum brake power. With the higher compression ratio, the NO emission for UTO is increased due to increase in-cylinder temperature [51].

4.6.5 Smoke opacity

Smoke is higher when a fuel's ratio of hydrogen to carbon is less than two [71]. Fig. 4.30 compares the smoke opacity of UTO and diesel at different brake power. It can be observed that the smoke opacity increases with an increase in the brake power as expected, but the smoke opacity of UTO is 7% lesser than that of diesel at maximum brake power. Lower CR of 16:1 shows a higher smoke opacity of 5%, whereas for CR 17:1 and 18.5:1 decreases by 4% and 10% than that of diesel at full load. This may be due to the maximum temperature during the combustion increases and this, in turn, decreases the smoke opacity [72].

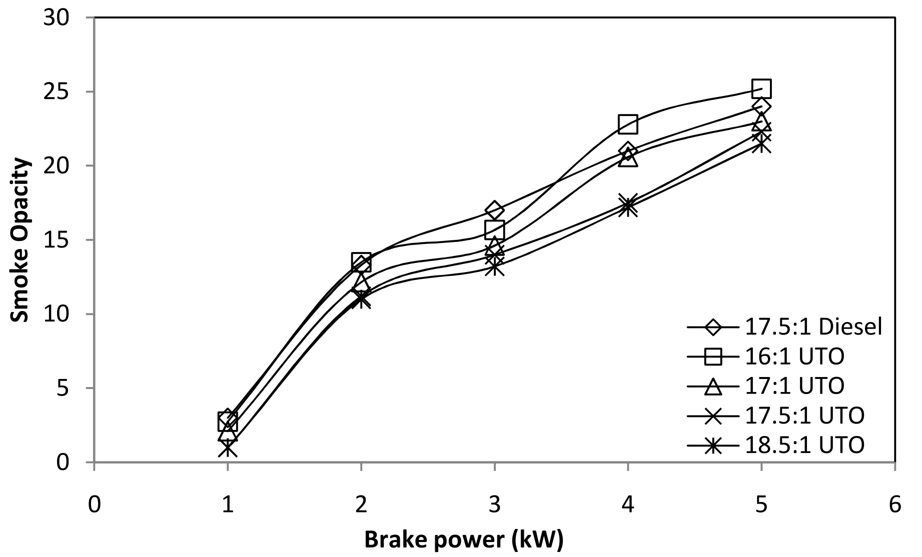

Fig. 4.30 Variation of smoke density with brake power

Phase III: Engine behavior studies of DI diesel engine operated at optimum injection timing of 20° bTDC and comparison between standard nozzle opening pressure of 200 bar and optimum nozzle opening pressure of 230 bar for different compression ratio.

The values of combustion, performance and emission parameters at full load are shown in table 4.1.

Table 4.1 Comparison table of combustion, performance and emission parameters

Parameters	Diesel	Nozzle opening pressure (200 bar) fueled with UTO				Nozzle opening pressure (230 bar) fueled with UTO			
CR	17.5:1	16:1	17:1	17.5:1	18.5:1	16:1	17:1	17.5:1	18.5:1
Ignition delay, (^{o}CA)	11.2	11.5	10.8	10.1	10	10	9.9	9.8	9.5
Max. HRR, (J/^{o}CA)	52.02	40.78	45.65	50.8	53.27	45.48	48.87	52.9	54.45
Peak cylinder pressure, (bar)	78.17	61.32	70.63	85.9	89.06	54.17	68.34	86.4	88.42
ROPR, (J/^{o}CA)	6.34	3.78	4.78	5.5	5.59	3.09	3.67	4.2	5.64
Combustion duration, (^{o}CA)	40.8	49.2	46.9	32.8	43.5	38.02	40.6	38.7	34.4
BTE, (%)	30.74	28.93	29.63	30.4	30.74	27.62	30.49	28	31.7
EGT, (^{o}C)	269.5	385	392.1	354.3	379.7	392.1	352.2	342	336.7
CO emission, (g/kWh)	0.005	0.013	0.012	0.012	0.011	0.013	0.012	0.002	0.010
CO_2 emission, (g/kWh)	7.7	2	1.9	1.8	1.7	1.9	2.3	2.7	2.8
HC emission, (g/kWh)	3	8	7.3	6.92	6	8.13	6	4.3	4
NO emission, (g/kWh)	1252	1174	1407	1597	1635	1500	1601	1625	1670
Smoke opacity, (m^{-1})	24	26.4	25.2	25.5	23	25.2	23	22.3	21.5

CHAPTER 5

CONCLUSION

5.1 Conclusion

The combustion, performance and emission characteristics of a single cylinder, four stroke, air cooled, direct injection diesel engine having a power output of 4.4 kW at a constant speed of 1500 rpm, fueled with UTO, diesel blends and diesel have been analyzed and compared with those of diesel. The following are conclusions；

- ✓ The UTO can be used as a fuel in the CI engines as it possesses a heating value. Considering the specific energy consumption, UTO with CR18.5 can be the optimum CR tested.
- ✓ The ignition delay for the UTO is shorter by about 1-3 °CA compared to that of diesel in the entire range of operation.
- ✓ The HC and CO emissions for the all CR of UTO are marginally higher than those of diesel operation at full load.
- ✓ The NO emission is higher at optimum CR 18.5 for UTO fuel than diesel at full load.
- ✓ Smoke is lower with the UTO than diesel at full load. The smoke value of UTO is lower at CR 18.5 than that of diesel at full load.

5.2 Future Scopes

To fulfill the demand of energy, various alternative fuels have been developed to replace fossil fuel. These alternative fuels have to attain a suitable fuel property by proper treatment of fuel. As per the experiment, used transformer oil (UTO) has been proved as an alternate substitute for the diesel engine.

In the present investigation, higher engine efficiency was observed with the higher compression ratio. Along with higher efficiency, an increase in the NO emission was also observed which is very harmful and toxic exhaust emission, but other emissions of engine were found to be lower with higher compression ratio. Thus to use UTO as an alternative fuel, NO emission should be reduced as much as possible. This can be reduced by reducing the peak cylinder temperature.

Methods to reduce NO emissions are;

- o Water / ethanol injection
- o Exhaust gas recirculation
- o Duel fuel operation

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ANNEXURE

1. Details of instruments

Si. No.	Instrument	Measurement	Make and model	Measurement technique / method
1	Load cell	Loading device	Indian make	Load cell
2	Burette	Fuel consumption	Indian make	Solenoid type
3	Temperature indicator	Exhaust gas measurement	Indian make	Thermocouple
4	Exhaust gas analyzer	NO	AVL 444 Digas analyzer	Chemiluminescence
		HC		FID
		CO		NDIR
5	Pressure transducer with charge amplifier	Cylinder pressure	Kistler 5395A	Piezoelectric pickup
6	Crank angle encoder	Crank angle	Indian make	Magnetic pick up type

2. Range, Accuracy and Uncertainity

Si. No.	Instrument	Range	Accuracy	Uncertainty
1	Load indicator	250 - 6,000 W	±1 W	0.2
2	Temperature indicator	0 - 900	±1 ^{o}C	0.15
3	Burette	1- 30cc	±0.2 cc	1.5
4	Speed sensor	0 - 10,000 rpm	±10rpm	±1
5	Exhaust gas analyser	NO (0 - 5,000 ppm)	±50ppm	1
		HC (0 - 200,000 ppm)	±10ppm	0.5
		CO (0-10%)	0.03%	1
6	Smoke meter	0-100%	±1 %	1
7	Pressure transducer	0-110bar	±1bar	0.15
8	Crank angle encoder		±1	1

3. Technical specification of smoke meter

Parameter	OPACITY	ABSORPTION	RPM	Oil Temperature
Measuring range	0-100 %	0-99.99 m^{-1}	400-6000 min^{-1}	0-150^{o}C
Accuracy & Repeatability	± 1 % of full scale	Better than ± 0.1 m^{-1}	± 10	± 3^{o}C
Resolution	0.10%	0.01 m^{-1}	± 1	± 1^{o}C

4. *Technical specification of exhaust gas analyser*

Application	: For free-acceleration test only	
Calibration	: Automatic (self-calibration immediately after switch-on or at the press of a key).	
Display	: Digital	
Standard	: LED (7 segment), 4x15 (mm)	
Optional	: LCD	
Alarming signal	: Equipment is not working.	
Linearity check	: (48.4% - 53.1%) / (1.54m^{-1} - 1.76m^{-1}) of measurement range (manual).	
Probes	: Set of probes of three different size (10, 16 &27 mm internal diameter)	
Hose pipes	: material	length
Standard	Rubber	4.0 meter
Optional	Rubber	2.5/5.0 meter
Smoke inlet	: Through a control valve.	
Smoke temperature at entrance	: 250^{o}C (maximum)	
Measuring chamber		
Length	: 430±5 mm	
Heating	: Thermostatically controlled.	
Light source	: Halogen Lamp, 12 V/5W	
	(Color temperature: 3000±150 K)	
Sensor	: Selenium Photocell (size – ϕ 45 mm) 65	
Built-in printer	: Dot matrix printer, 24 column	
Ambient operating conditions		
Temperature	: 0-50^{o}C	
Humidity	: 90% at 50^{o}C (non-condensing)	
Dimension	: width x height x depth	
Basic unit 437C	: 600 x 260 x 370 mm	
RPM module	: 93 x 140 x 33 mm	
Dispeed 490	: 277 x 48 x 184 mm	
Weight		
Unit	: 24 kg	
RPM module	: 0.45 kg	
Dispeed 490	: 1.18 kg	
Hose pipe (4m) + probes	: 8 kg	
Power consumption		
Overall equipment	: 600 W	
Measuring chamber heating	: 500 W (at 220V)	
Storage temperature	: -30^{o}C to +65^{o}C	
Protection type	: IP 52	

Conference papers

1. **S.B. Ekka**, P. Behera and S. Murugan, *"Performance and emission studies of a diesel engine with different compression ratios"*, Proceedings of the International Conference on Alternative Fuels for I. C. Engines (ICAFICE), February 6-8, 2013, MNIT Jaipur.

Journal papers

1. P. Behera, **S.B. Ekka** and S. Murugan, *"Studies on the effect of variable compression ratio on the engine behavior fueled with used transformer oil"*, International Journal of Environment and Waste Management. (Under Review)
2. **S.B. Ekka**, P. Behera and S. Murugan, *"Studies on the effect of higher compression ratio on the engine behavior fueled with used transformer oil"*, International Journal Sustainable Energy. (Under Review)